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US 4969545 A

(58) Field of Search
UK CL (Edition P) F2D DCG, F2L LT
INT CL[6] F16D 48/06, F16H 61/14

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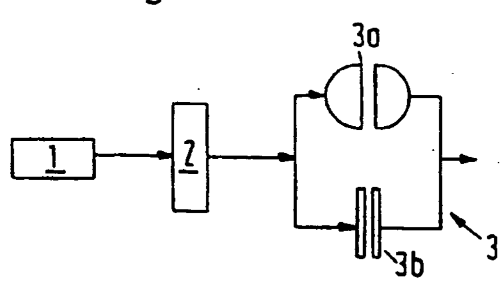
(54) Clutch control

(57) A method and device for controlling a torque transfer system with or without load distribution such as a torque converter, particularly for motor vehicles, wherein the clutch torque which can be transferred from a drive side to an output side of the torque transfer system is used as the control value. This control value is controlled by means of a setting member which is provided with a setting value which is functionally dependent on the transferable clutch torque, so that the transferable clutch torque always lies within a pre-determinable tolerance band about the slip limit. This slip limit is then accurately reached when the action of a torque arising on the drive side exceeds the clutch torque transferable by the torque-transferring parts.

Fig.1a



GB 2 319 817 A

Fig.1a

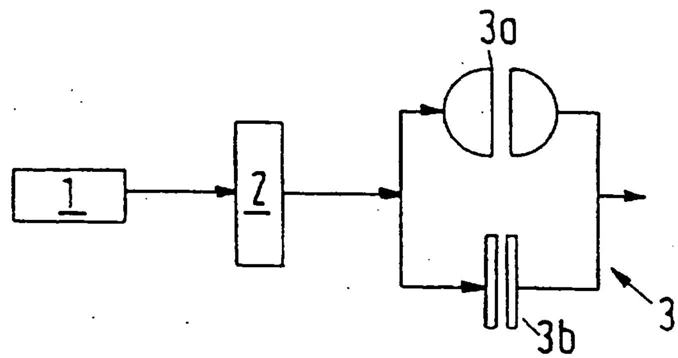


Fig.1b

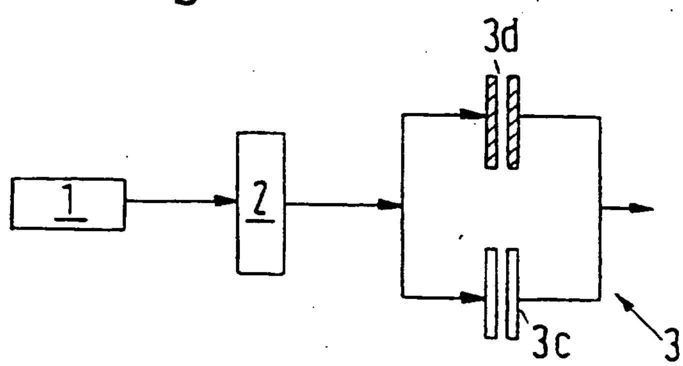


Fig. 2a

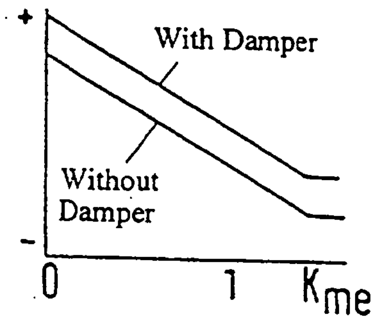


Fig. 2b

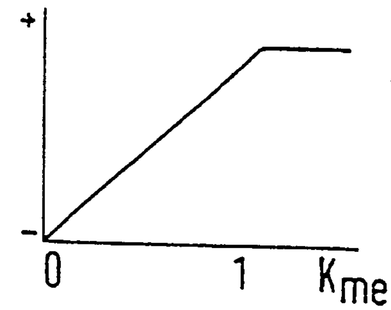


Fig. 2c

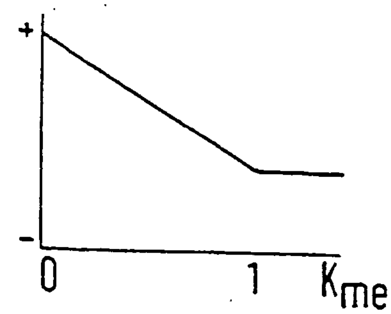


Fig. 2d

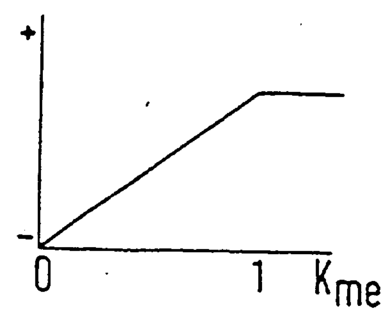
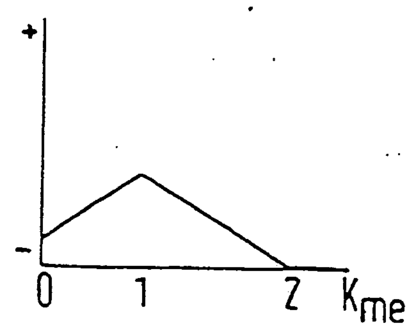
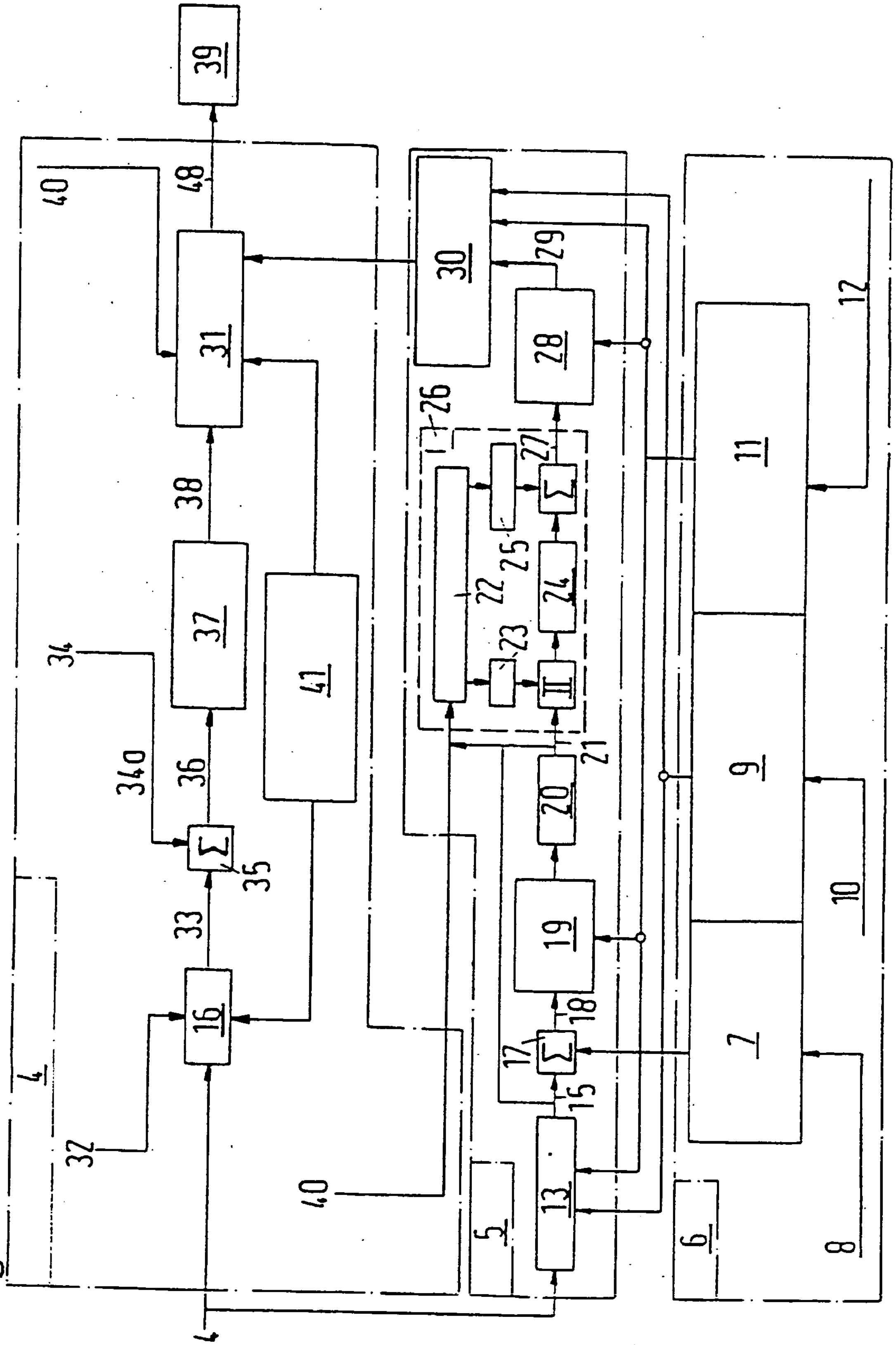


Fig. 2e

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48
40
31
30
29
28
38
26
27
22
25
24
23
37
41
34
34a
36
Σ
35
33
16
32
20
21
19
18
17
15
13
14
40
4
5
6
7
9
11
8
10
12
II
Σ

Fig.3

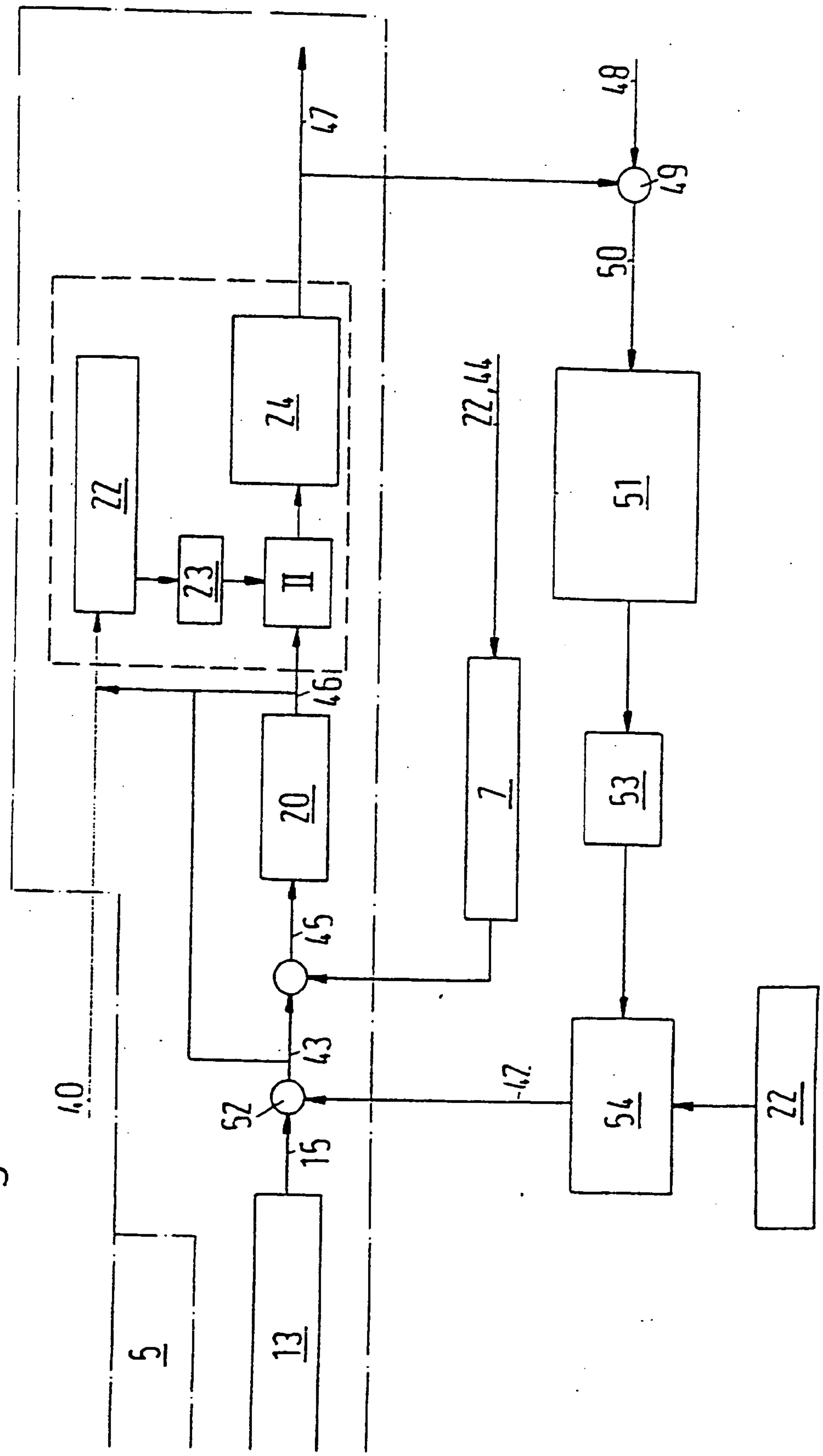


Fig.4

Fig.5a

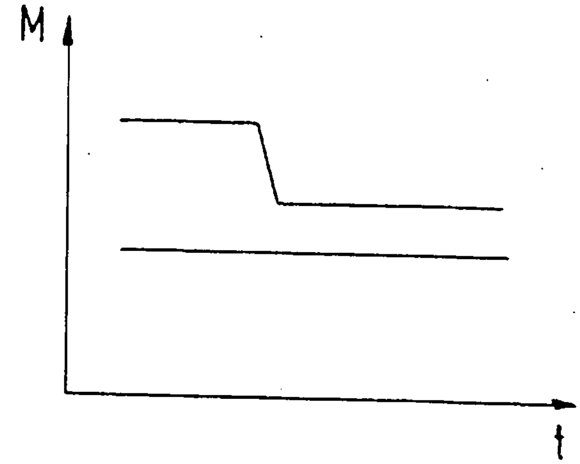


Fig.5b

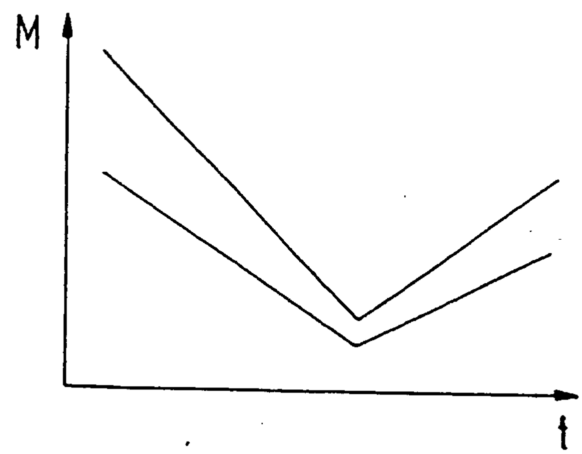


Fig.5c

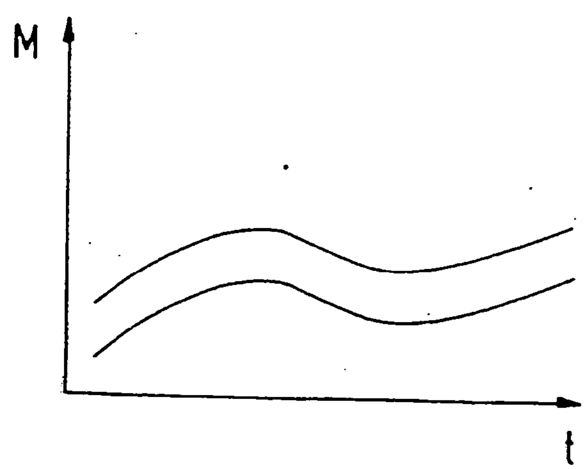


Fig.6

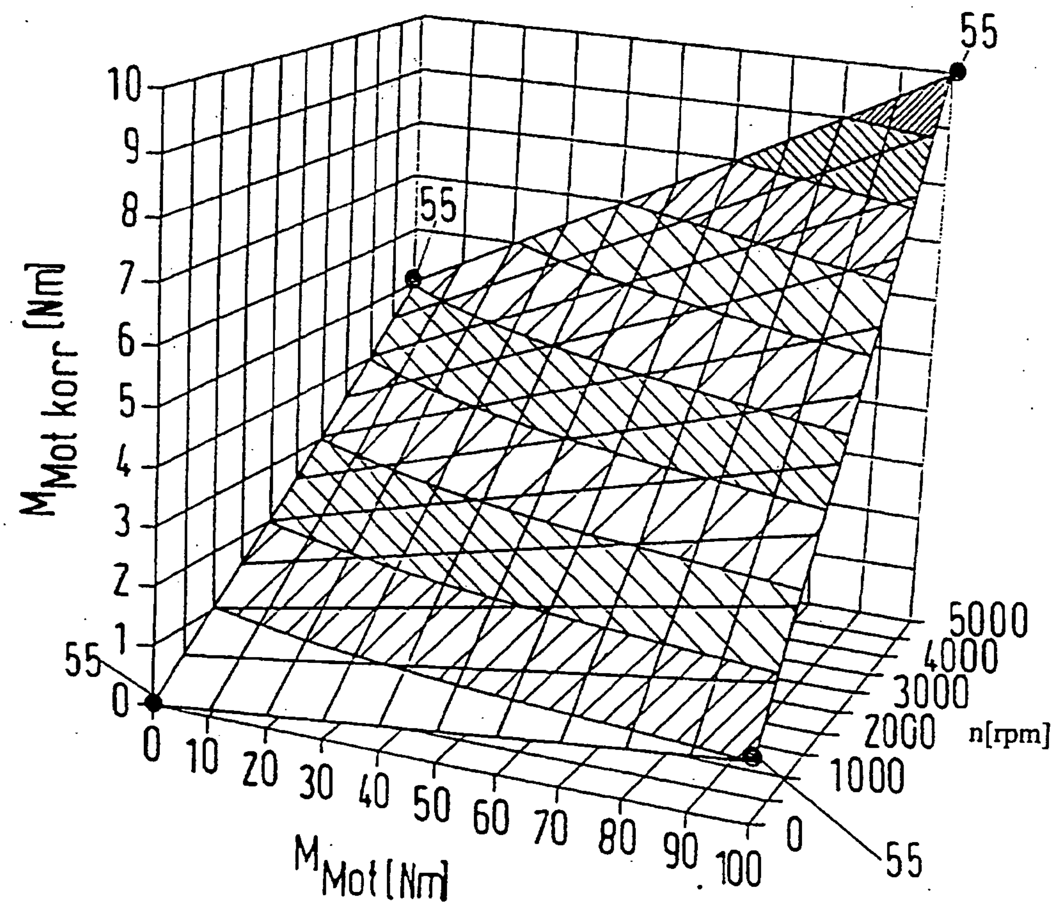

55
55
10
9
8
7
6
5
4
3
2
1
0
55
$M_{Mot\,korr}$ [Nm]
0
10
20
30
40
50
60
70
80
90
100
M_{Mot} [Nm]
5000
4000
3000
2000
1000
0
n[rpm]
55

Fig.6a

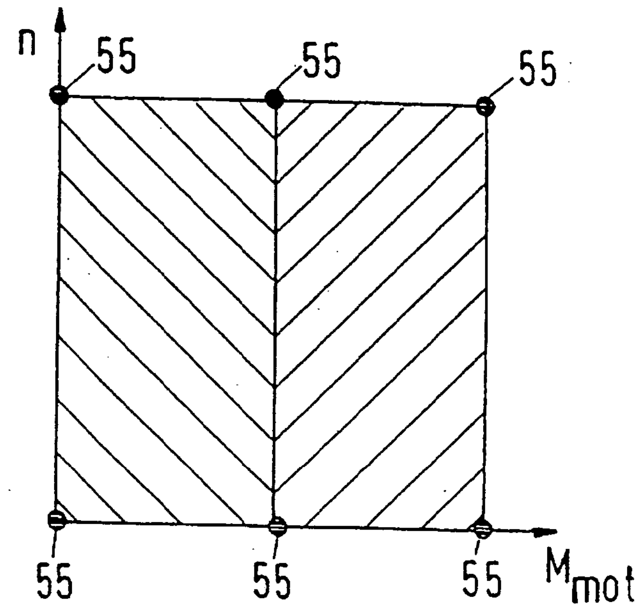


Fig.6b

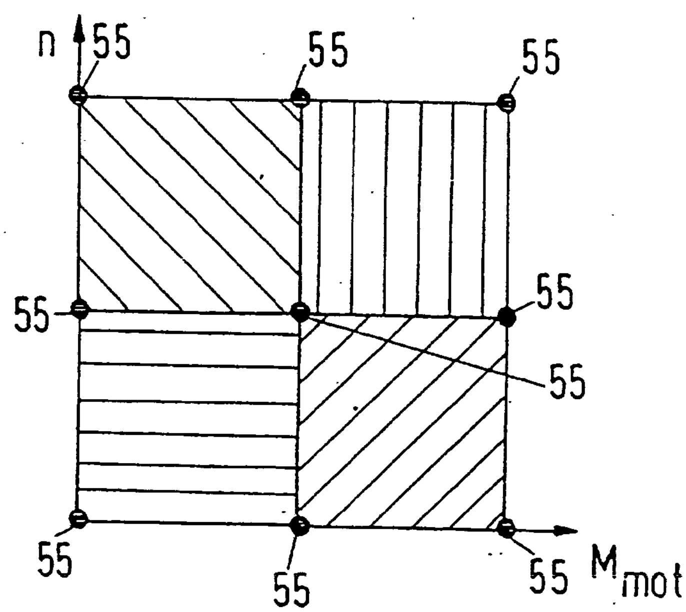


Fig.7

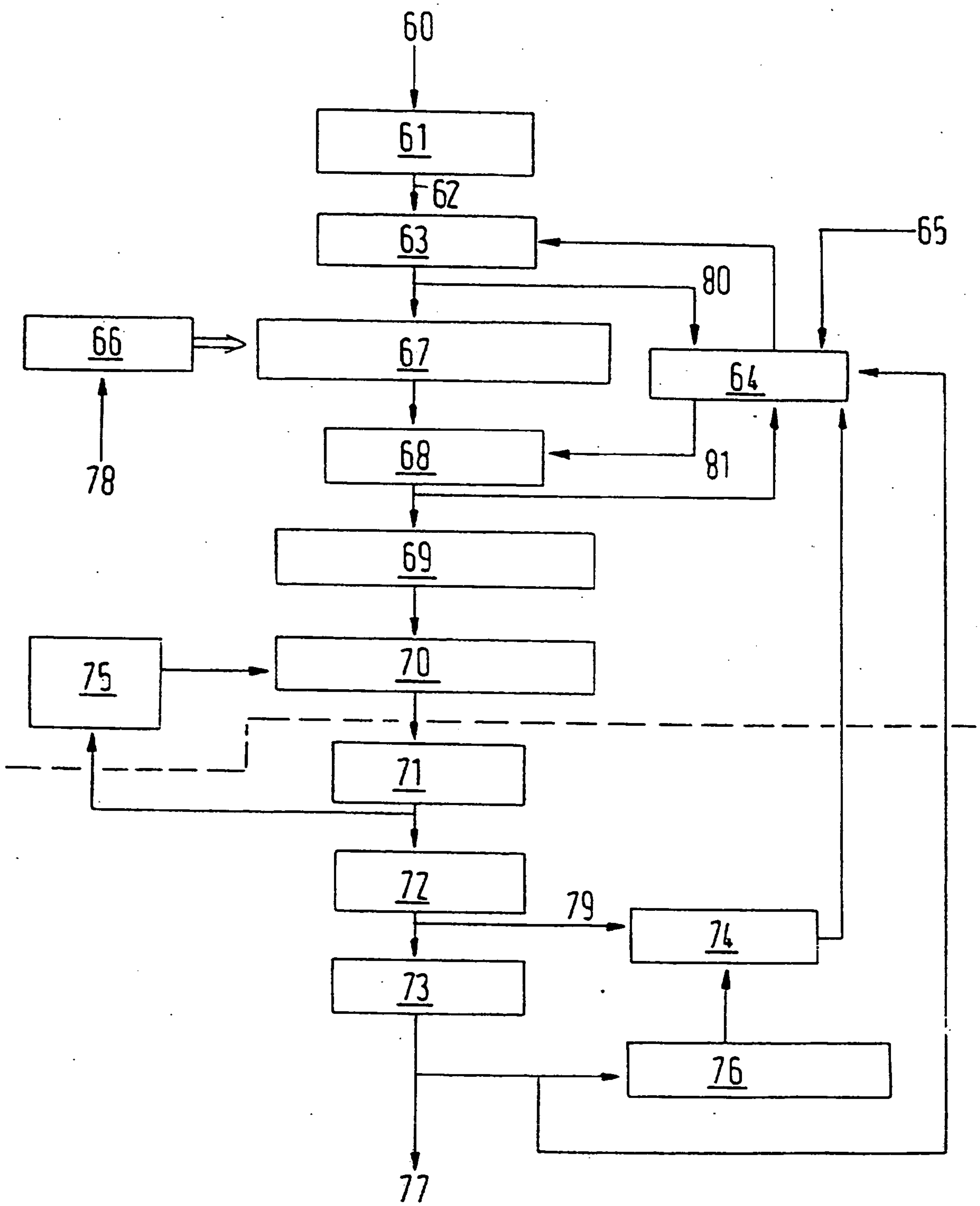


Fig.8

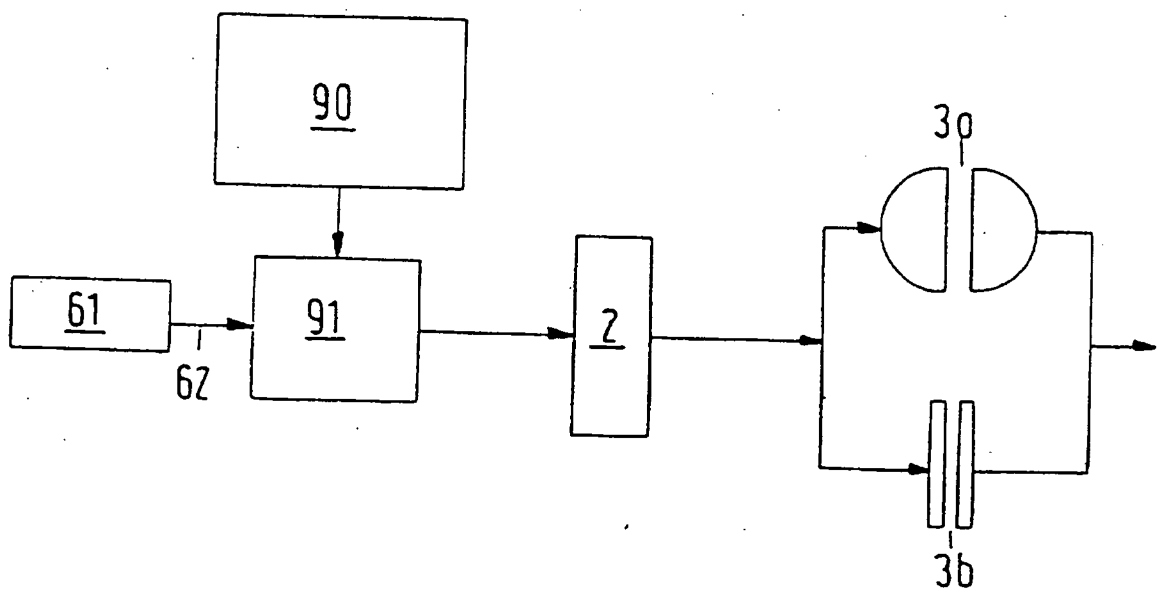


Fig. 9

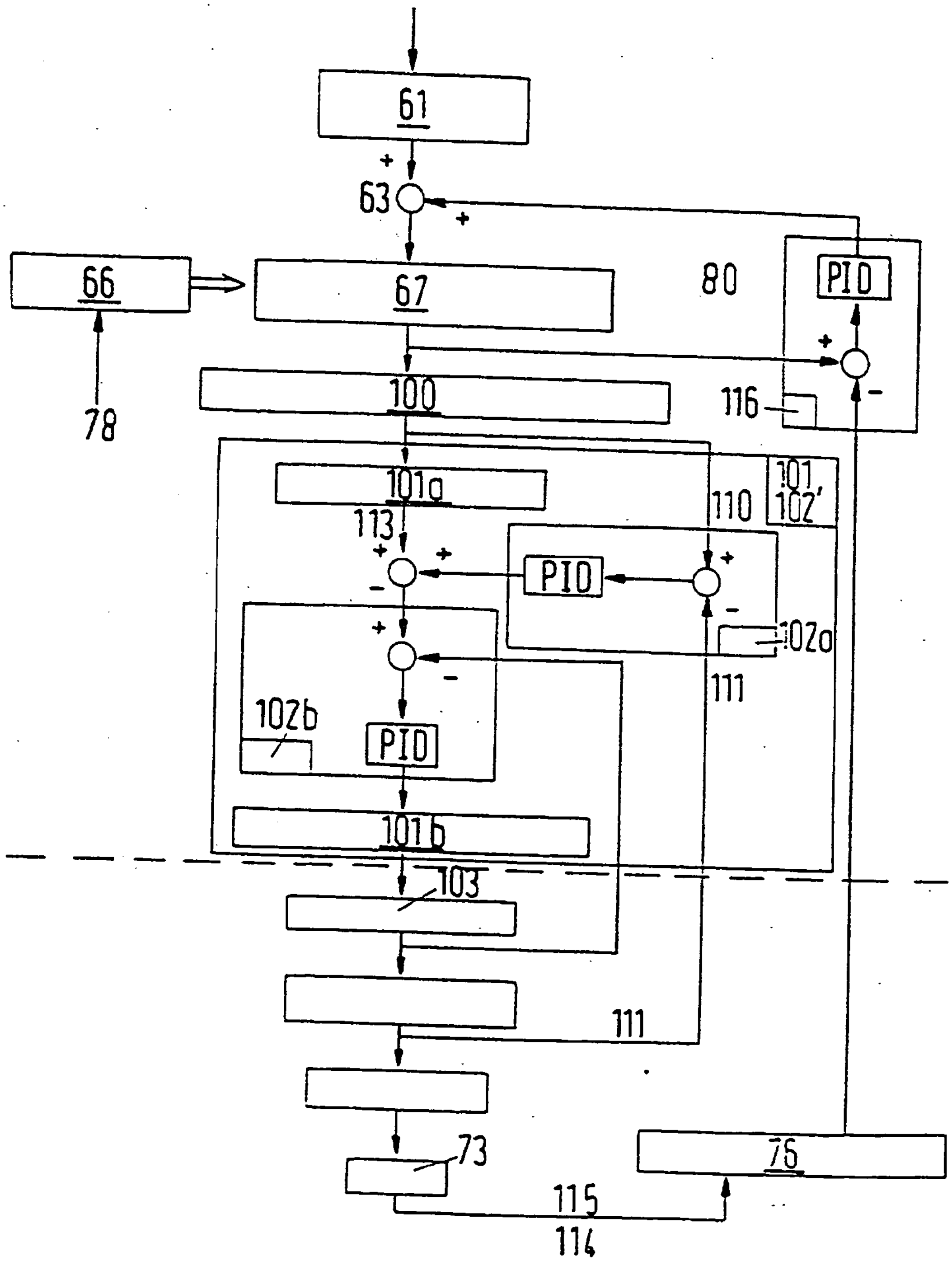

61
63
66
67
78
80
PID
116
100
101a
113
110
101, 102'
PID
102a
111
102b
PID
101b
103
111
73
76
115
114

Fig.10

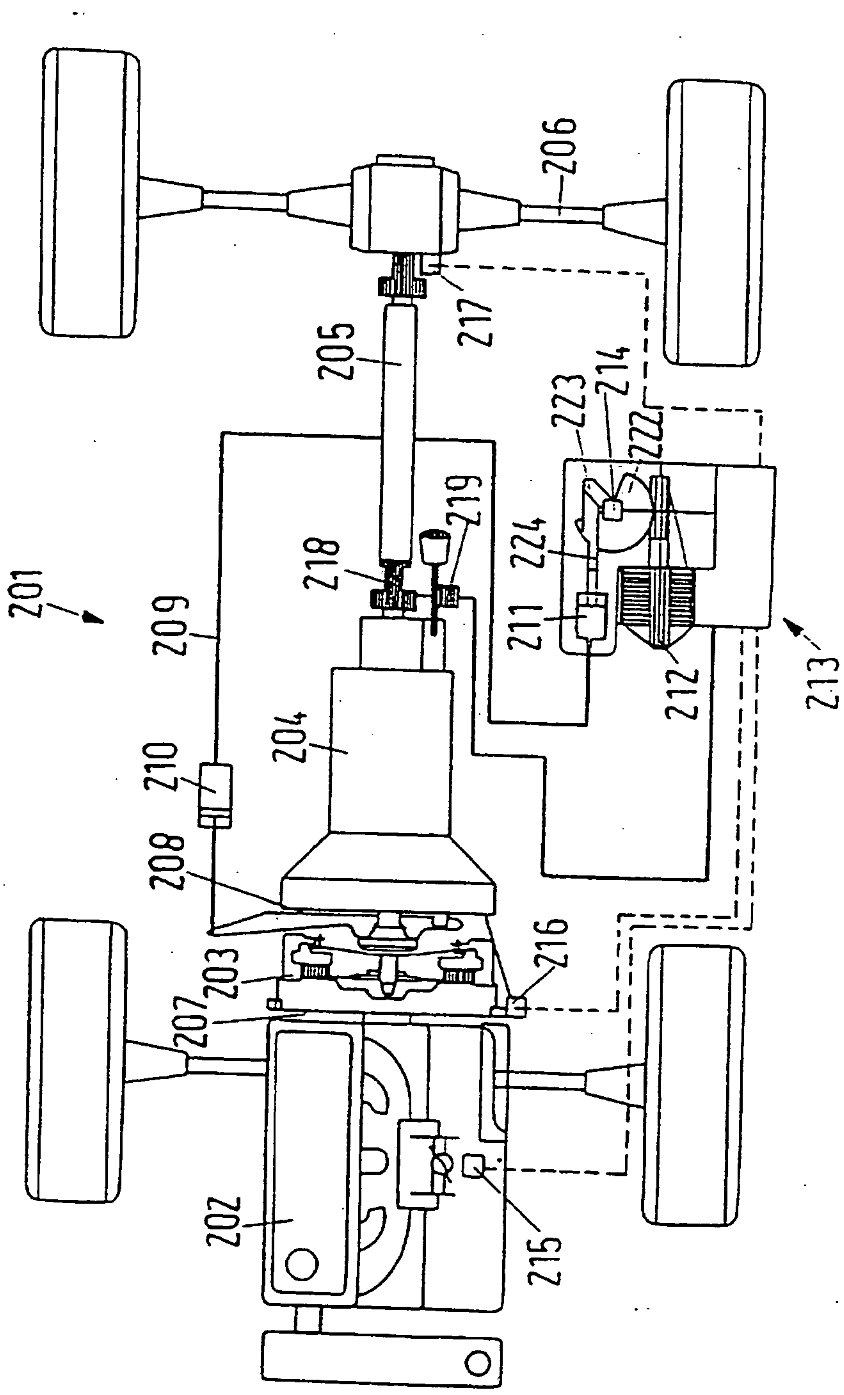

201
202
203
204
205
206
207
208
209
210
211
212
213
214
215
216
217
218
219
222
223
224

Fig. 11 a

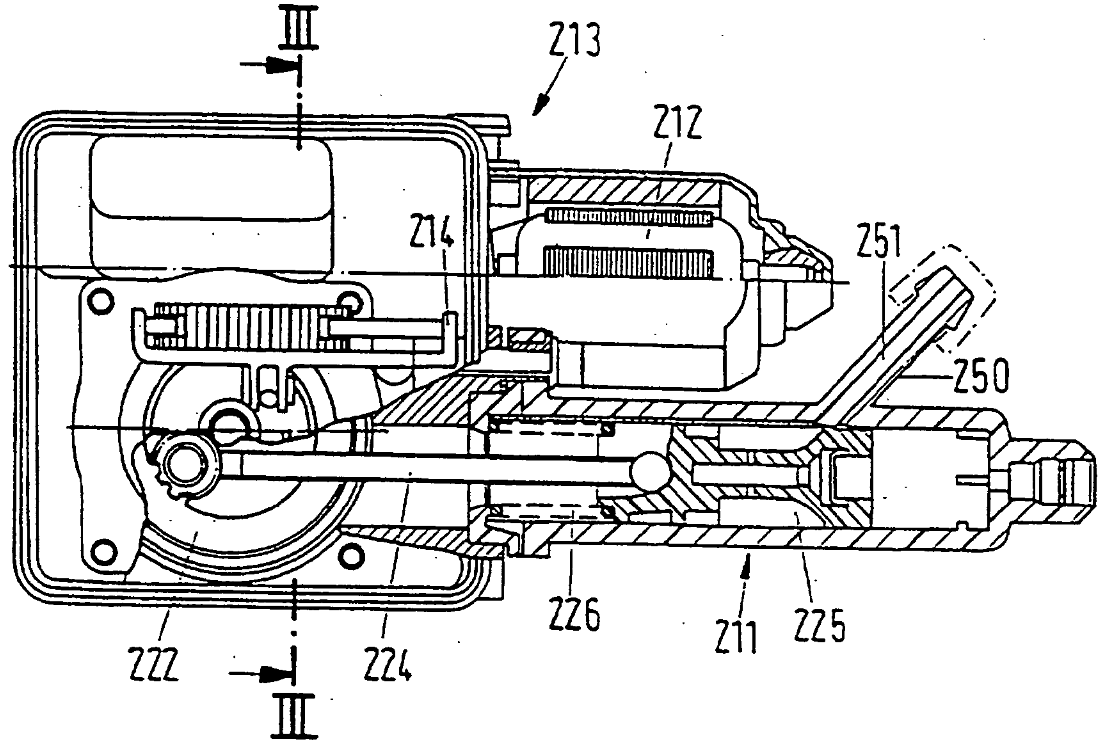
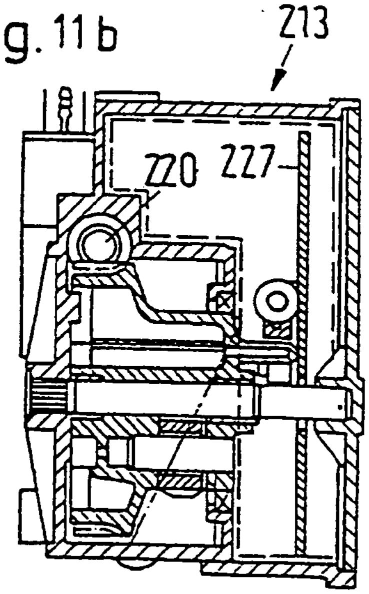


Fig. 11 b



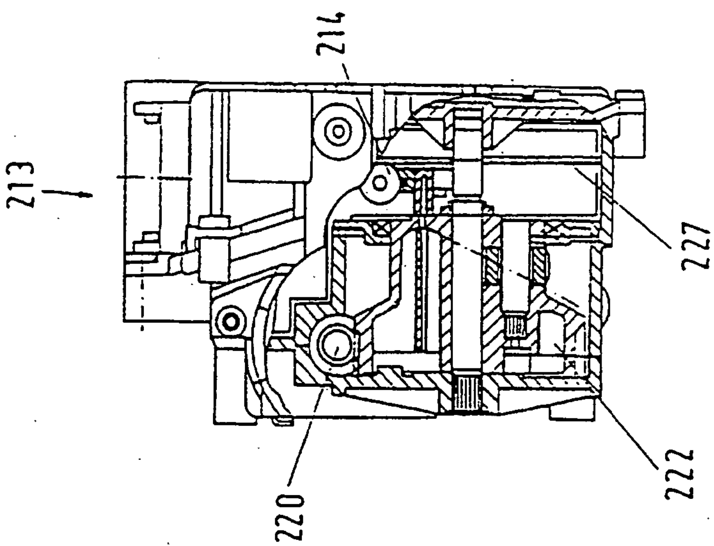


Fig.12b

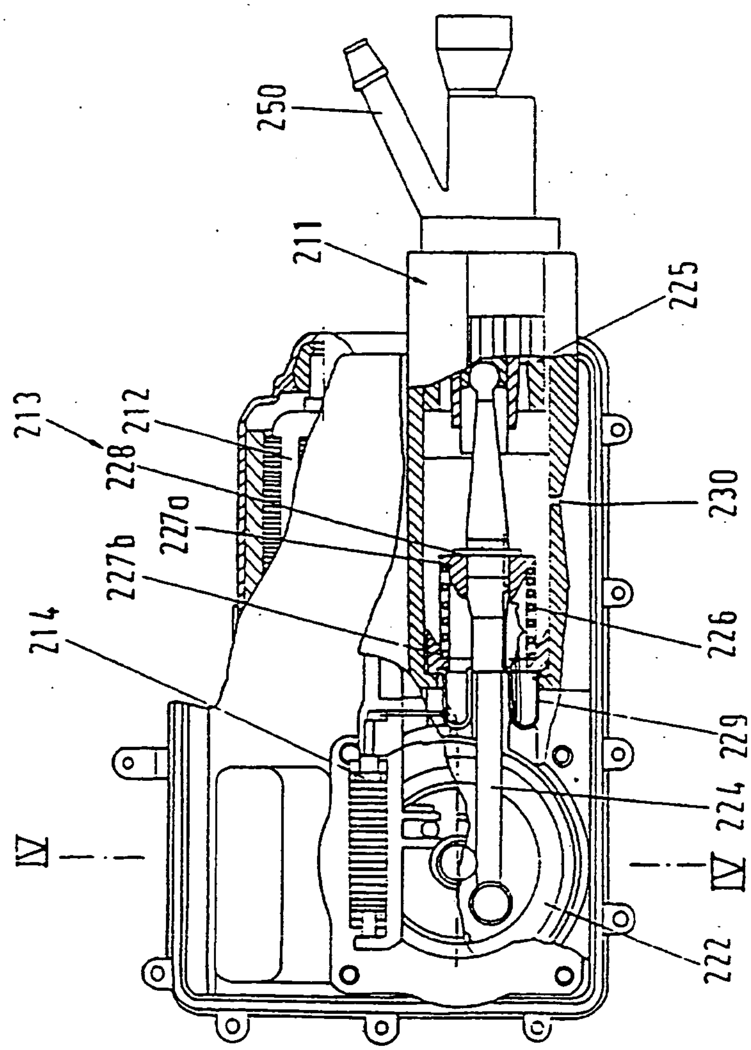


Fig.12a

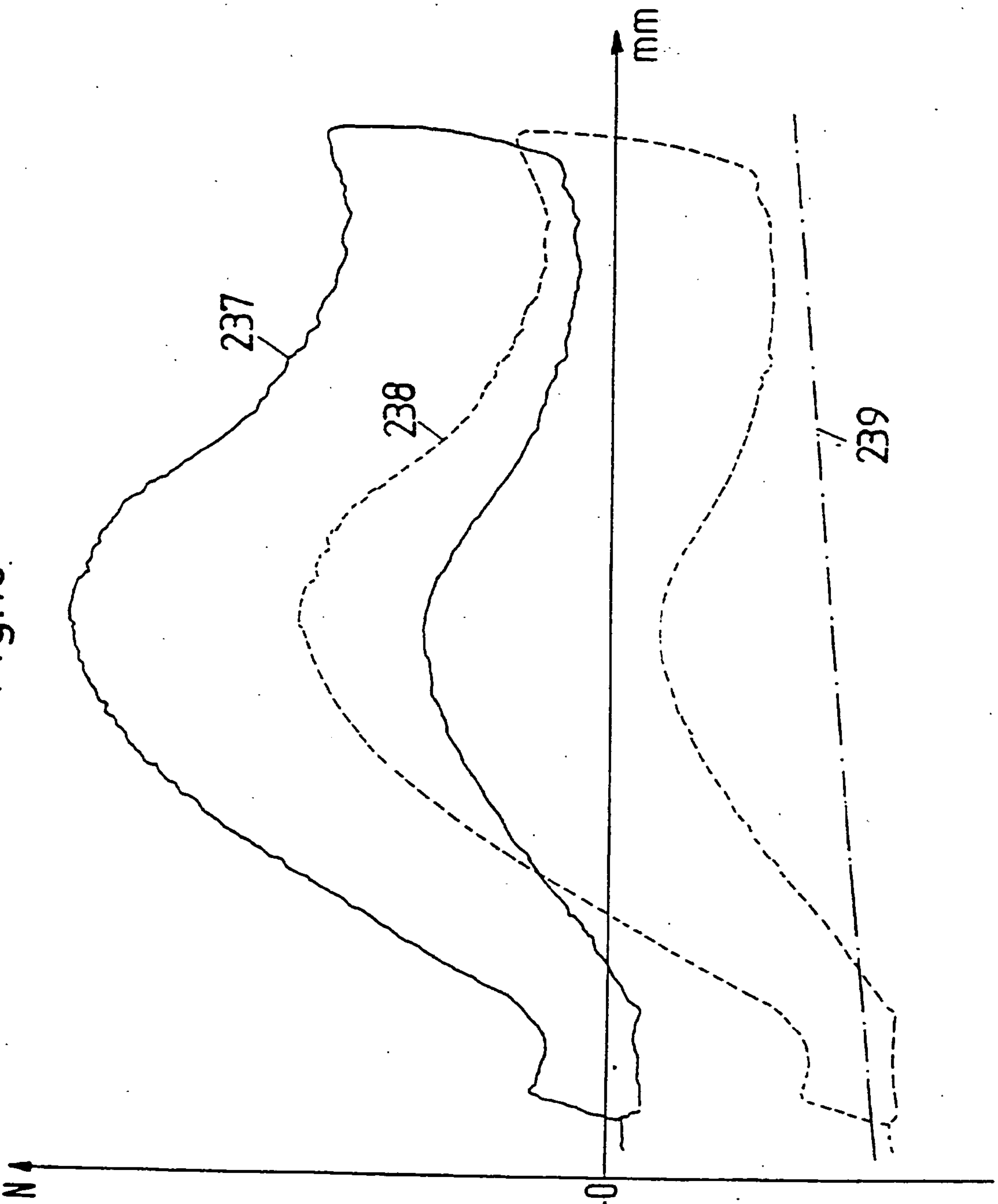


Fig.13

Fig.14

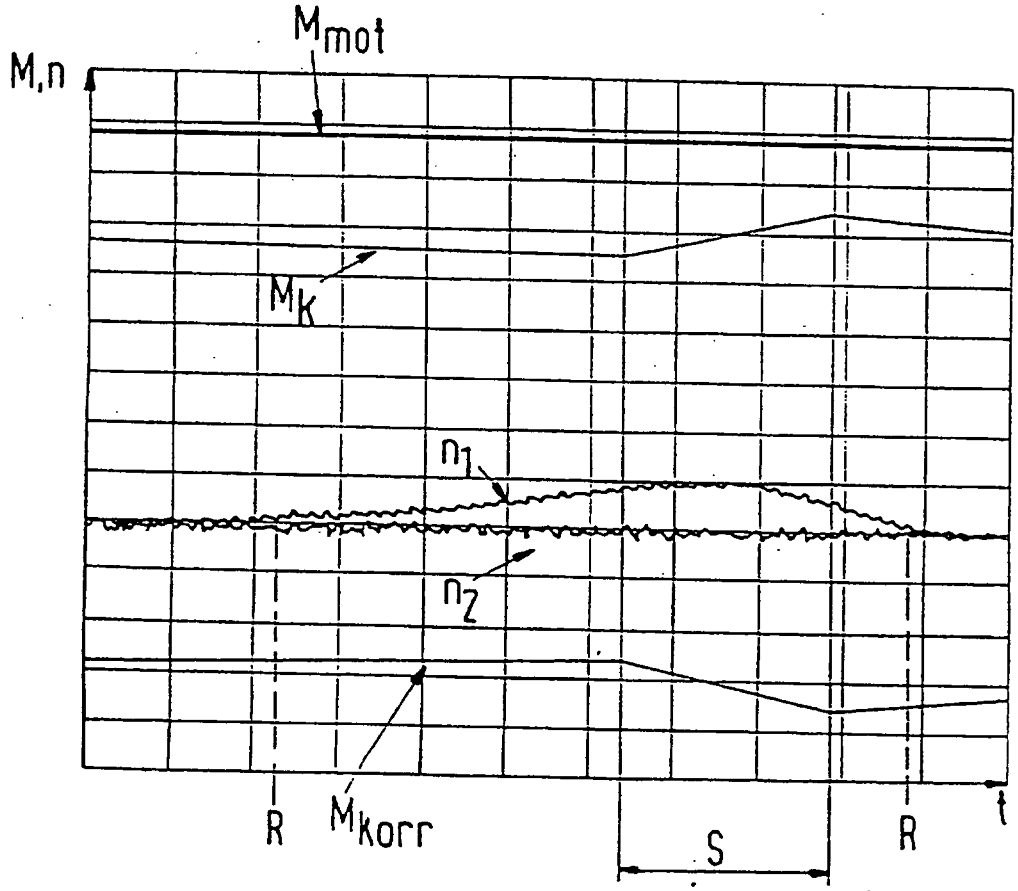


Fig.15

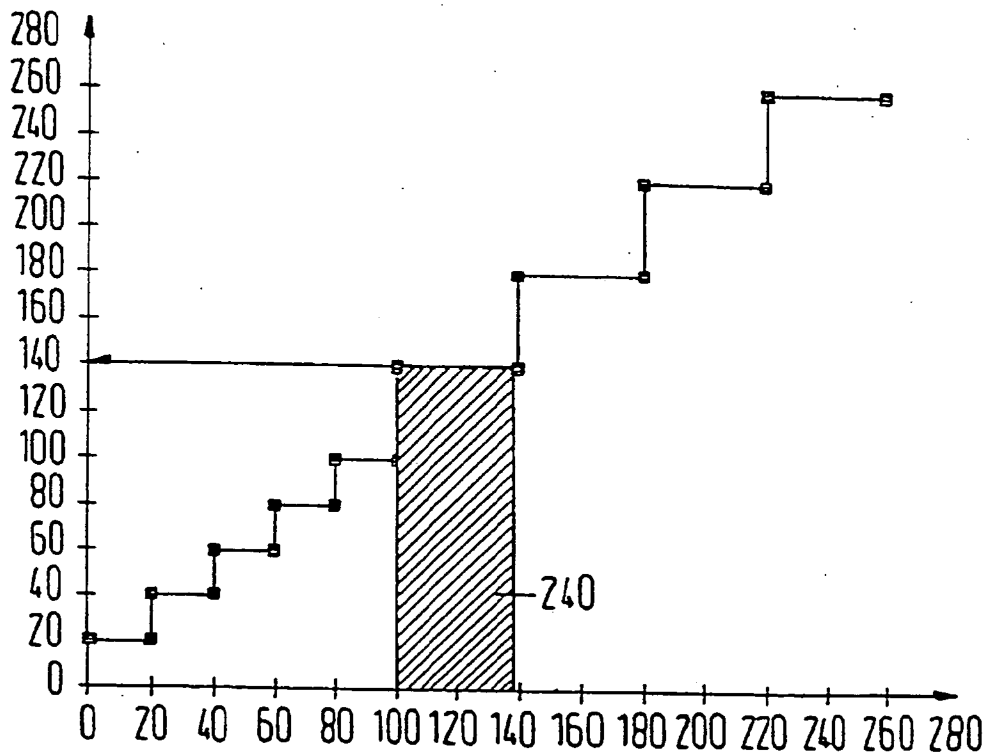

280
260
240
220
200
180
160
140
120
100
80
60
40
20
0
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280
240

Fig.15a

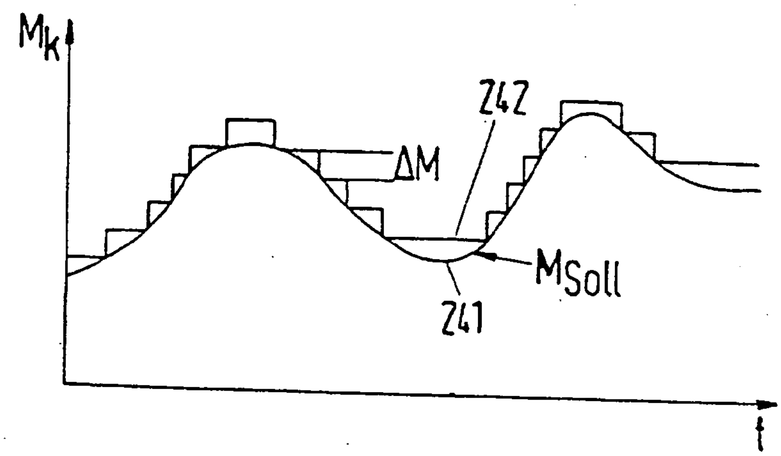


Fig.15b

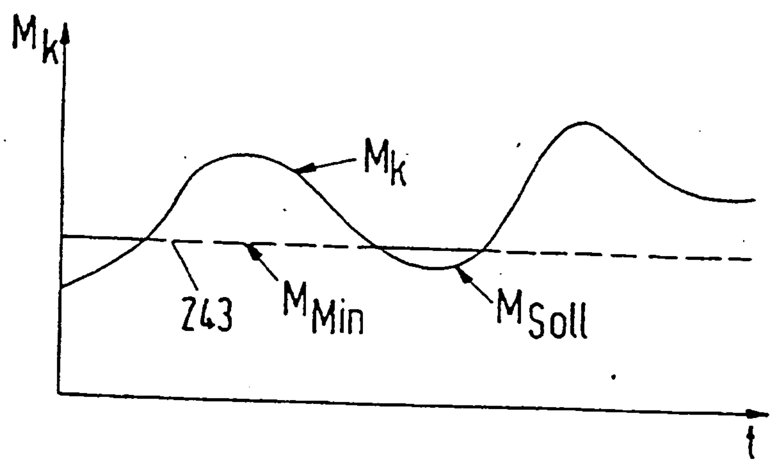


Fig.15c

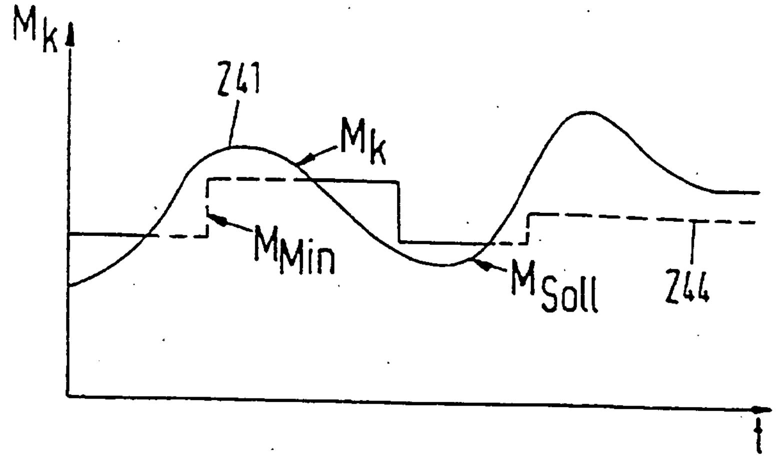


Fig.15d

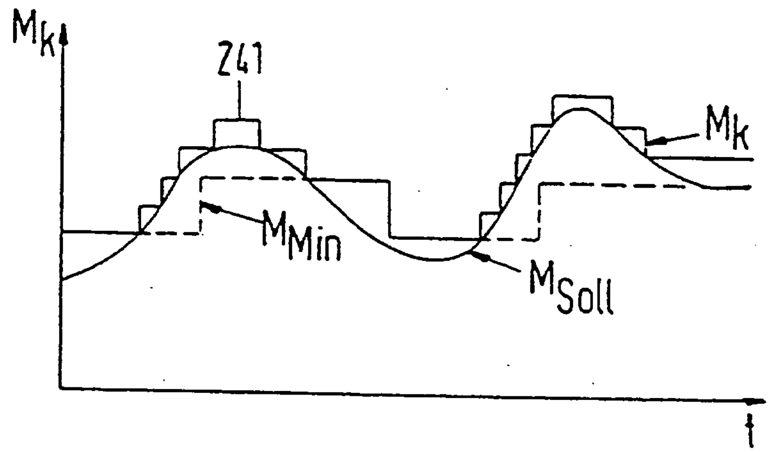


Fig.15e

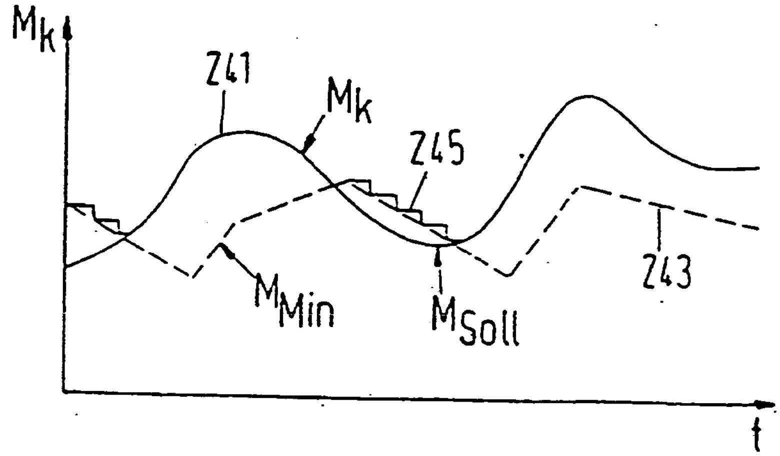
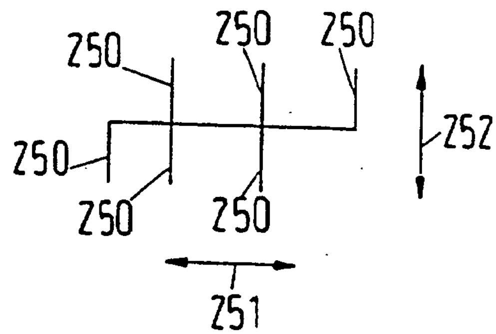
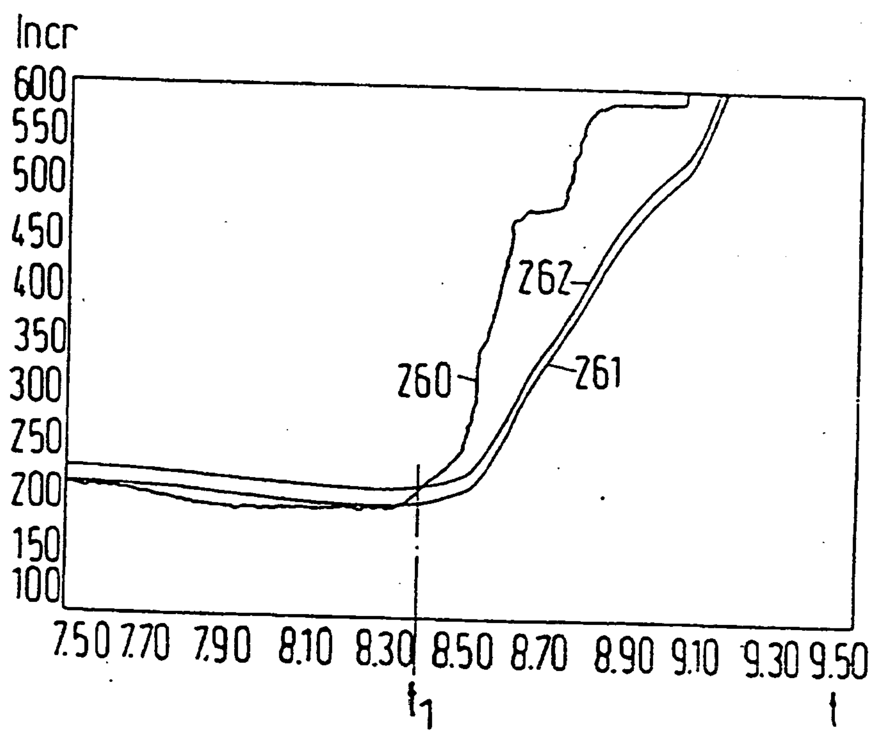


Fig.16



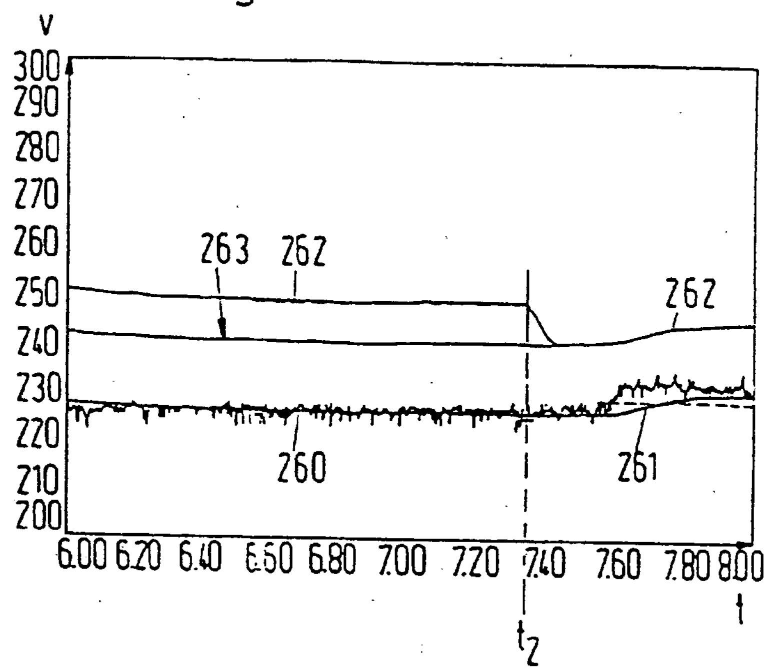
250
250
250
250
250
250
252
251

Fig.17



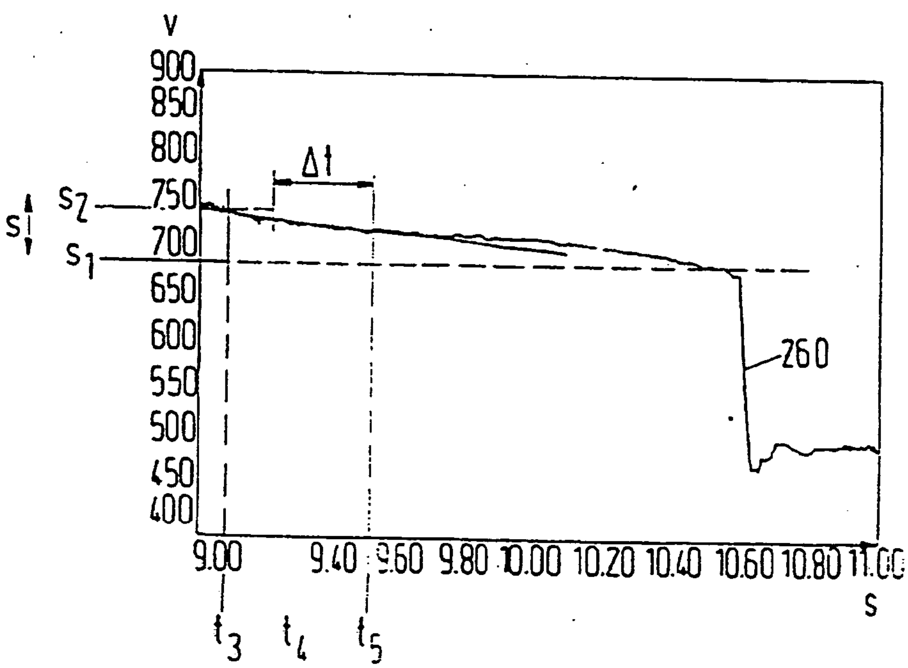
Incr
600
550
500
450
400
350
300
250
200
150
100
262
260
261
7.50 7.70 7.90 8.10 8.30 8.50 8.70 8.90 9.10 9.30 9.50
t_1
t

Fig.18



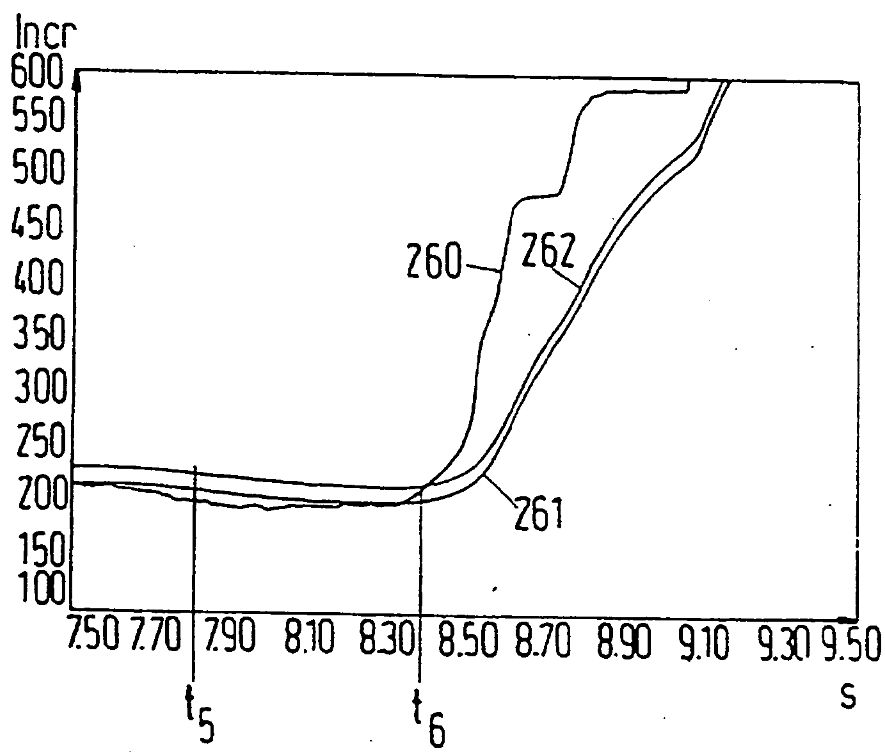
V
300
290
280
270
260
250
240
230
220
210
200
263
262
262
260
261
6.00
6.20
6.40
6.60
6.80
7.00
7.20
7.40
7.60
7.80
8.00
t
t_2

Fig.19



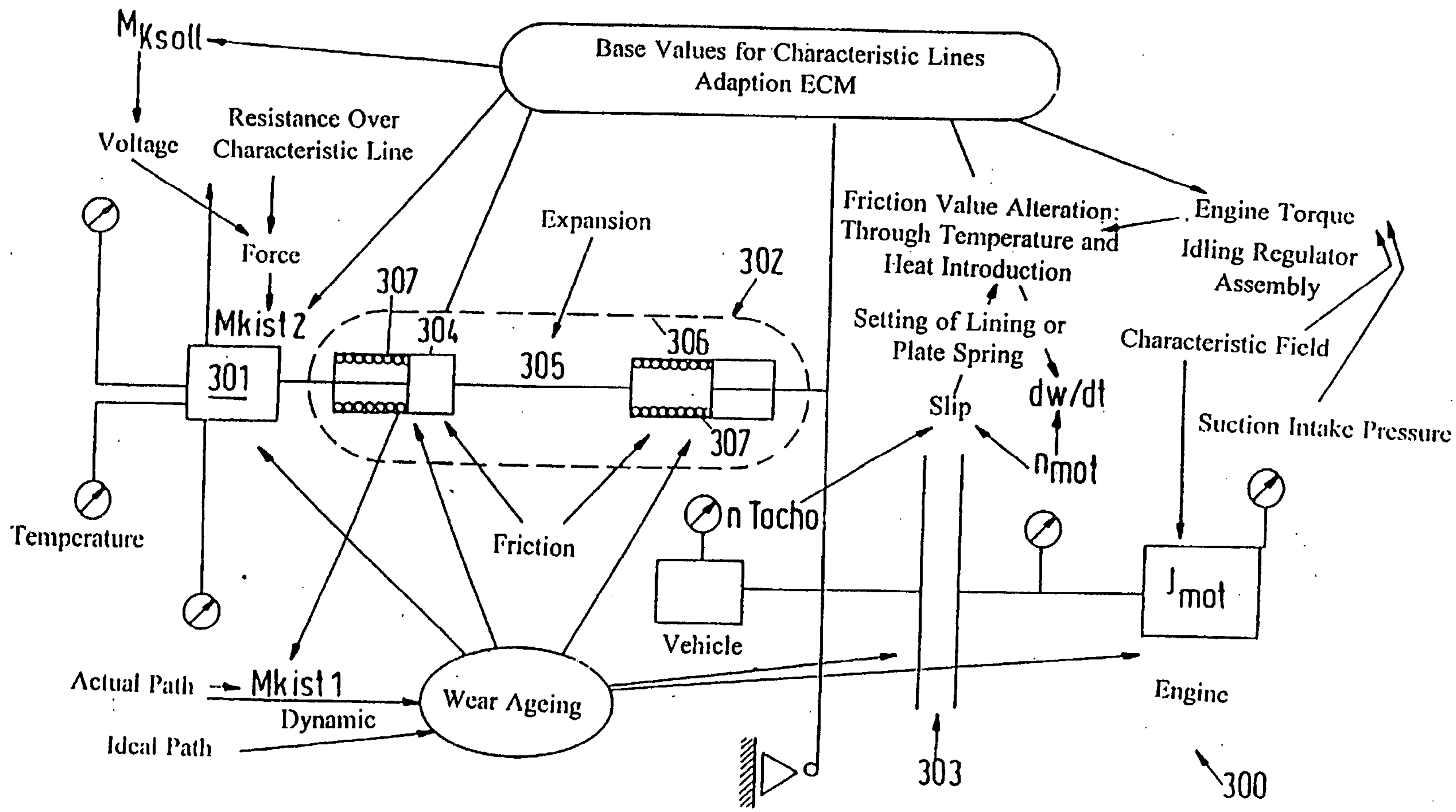
V
900
850
800
750
700
650
600
550
500
450
400
s_2
s_1
s
Δt
260
9.00
9.40
9.60
9.80
10.00
10.20
10.40
10.60
10.80
s
t_3
t_4
t_5

Fig.20



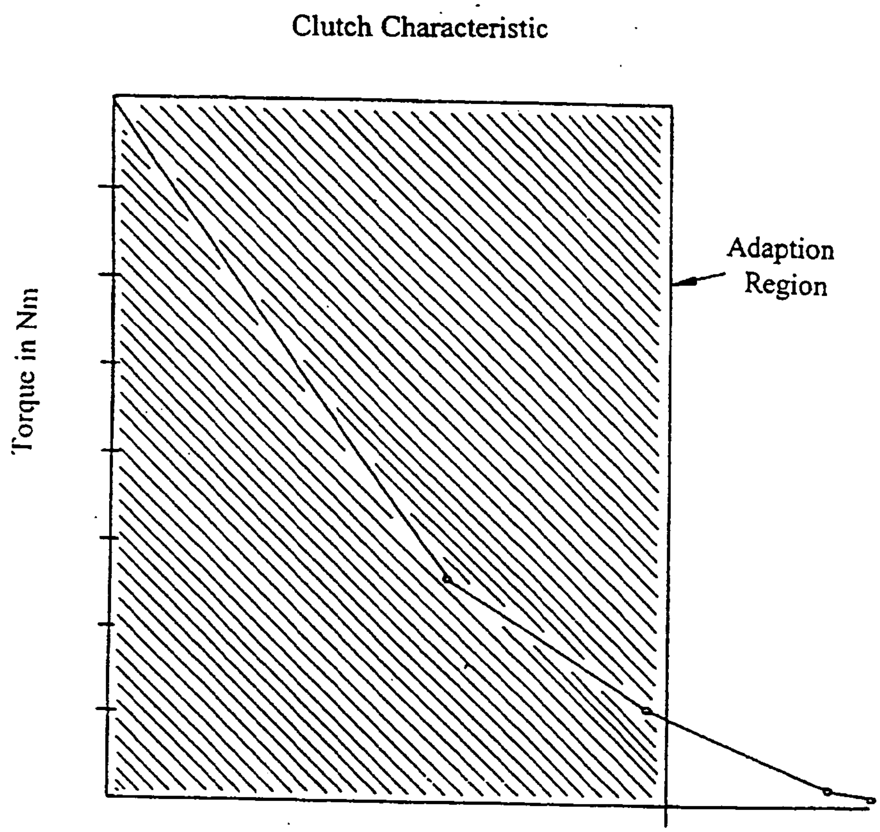
Incr
600
550
500
450
400
350
300
250
200
150
100
260
262
261
7.50
7.70
7.90
8.10
8.30
8.50
8.70
8.90
9.10
9.30
9.50
s
t_5
t_6

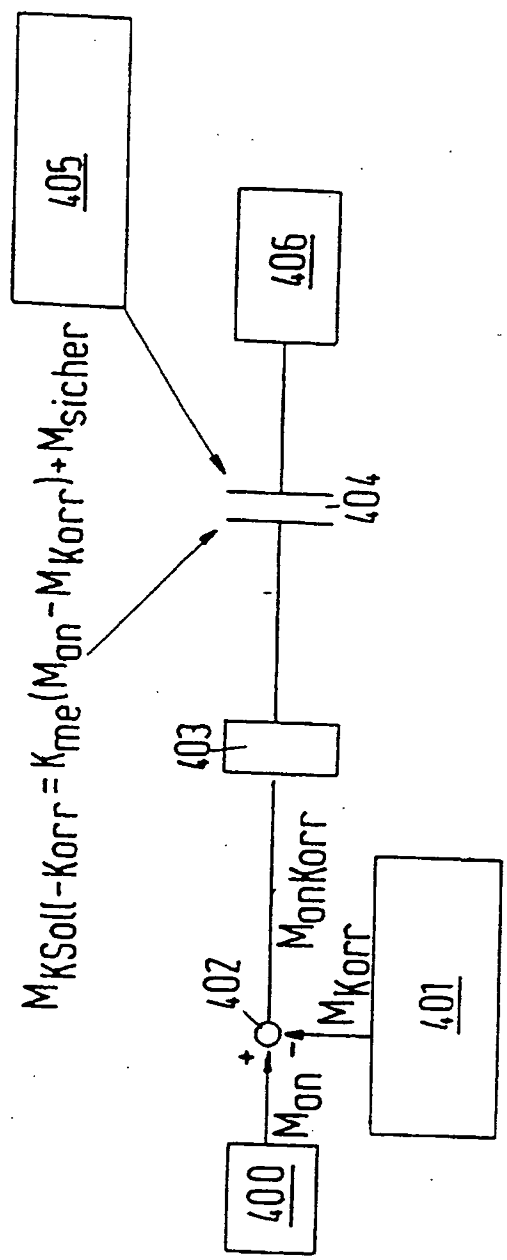
Fig.21



M_{Ksoll}
Base Values for Characteristic Lines Adaption ECM
Voltage
Resistance Over Characteristic Line
Force
Mkist 2
301
307
304
Expansion
305
302
306
307
Friction Value Alteration: Through Temperature and Heat Introduction
Engine Torque
Idling Regulator Assembly
Characteristic Field
Suction Intake Pressure
Setting of Lining or Plate Spring
Slip
dw/dt
n_{mot}
n Tacho
Temperature
Friction
Vehicle
J_{mot}
Actual Path
Mkist 1
Dynamic
Ideal Path
Wear Ageing
303
Engine
300

Fig.22

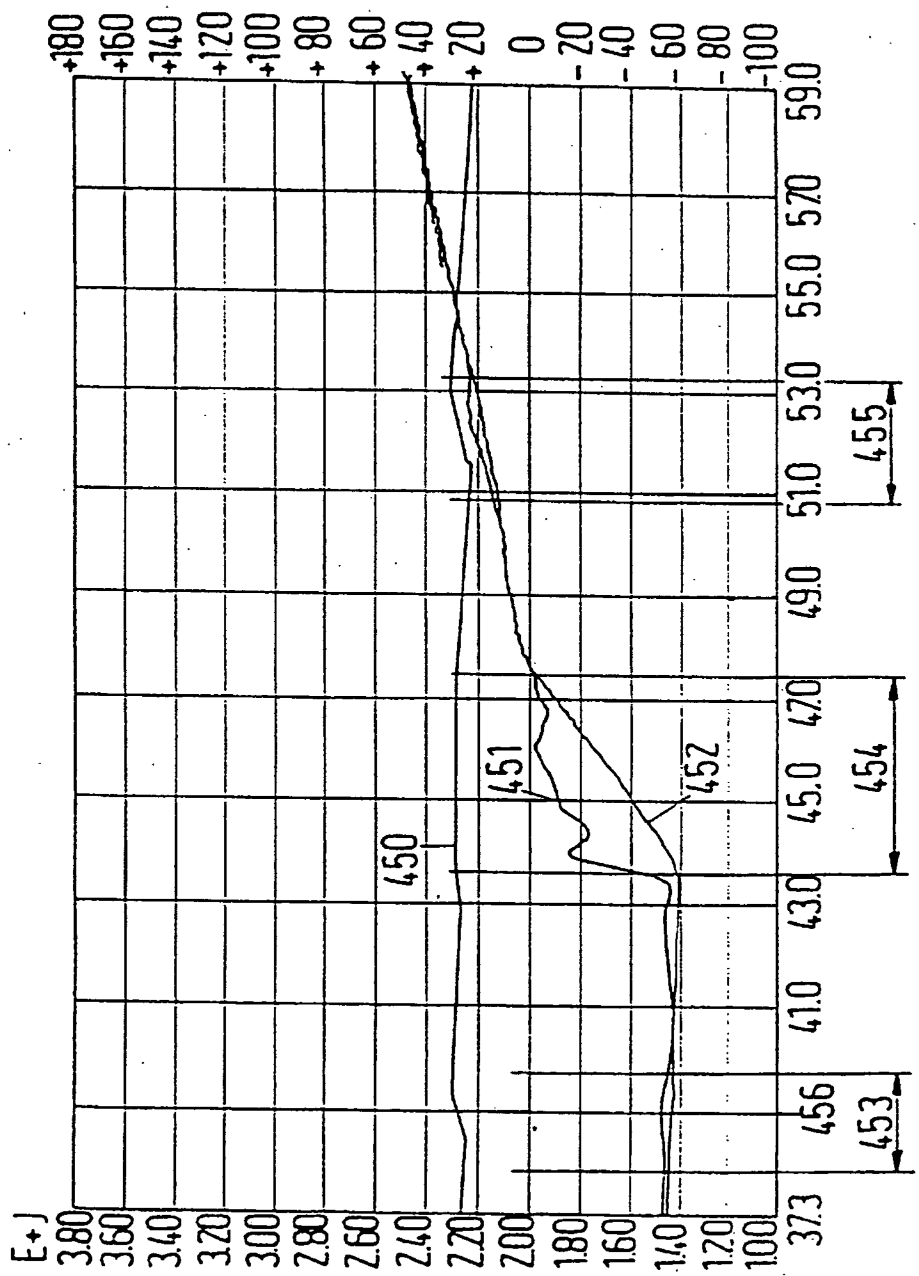




405
406
$M_{KSoll-Korr} = K_{me}(M_{an} - M_{Korr}) + M_{sicher}$
404
403
M_{anKorr}
402
M_{Korr}
401
+
-
M_{an}
400

Fig.23

Fig.24


E+J
3.80
3.60
3.40
3.20
3.00
2.80
2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
+180
+160
+140
+120
+100
+80
+60
+40
+20
0
-20
-40
-60
-80
-100
37.3
456
41.0
43.0
45.0
47.0
49.0
51.0
53.0
55.0
57.0
59.0
450
451
452
453
454
455

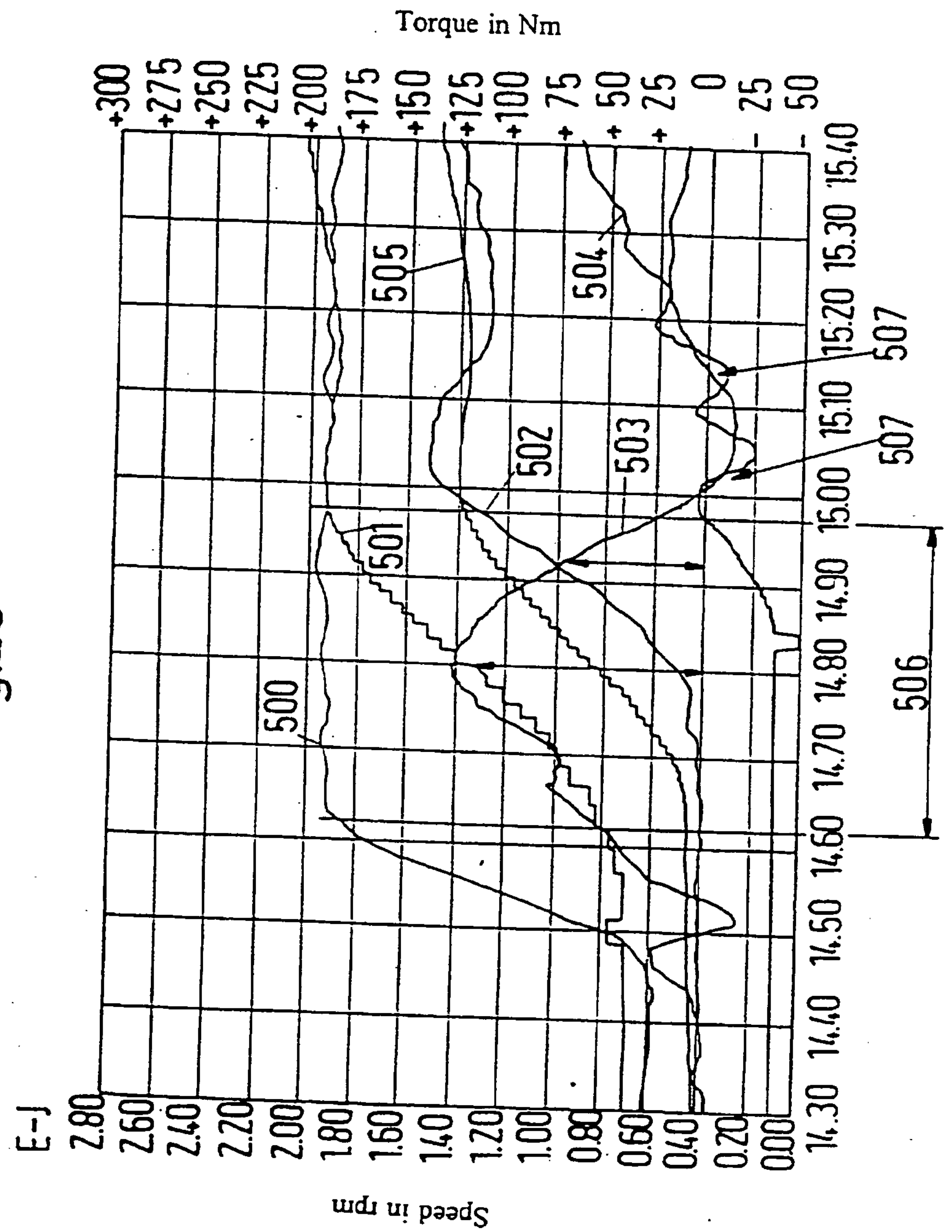
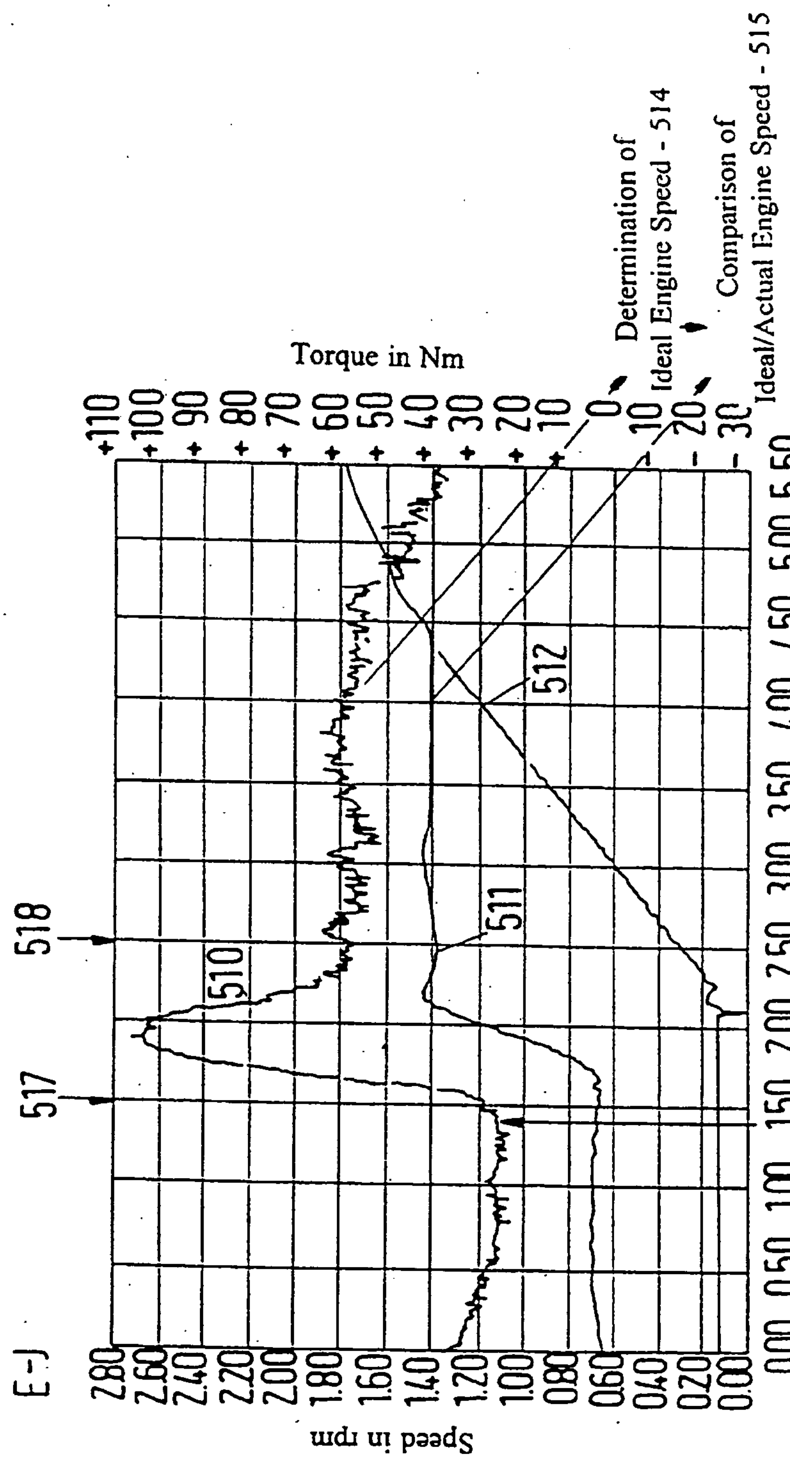

Torque in Nm
+300
+275
+250
+225
+200
+175
+150
+125
+100
+ 75
+ 50
+ 25
0
- 25
- 50
E-J
2.80
2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
0.80
0.60
0.40
0.20
0.00
14.30
14.40
14.50
14.60
14.70
14.80
14.90
15.00
15.10
15.20
15.30
15.40
Speed in rpm
500
501
502
503
504
505
506
507
507

Fig.25

Fig.26


Torque in Nm
Determination of Ideal Engine Speed - 514
Comparison of Ideal/Actual Engine Speed - 515
Speed in rpm
E-J
+110 +100 + 90 + 80 + 70 + 60 + 50 + 40 + 30 + 20 + 10 0 - 10 - 20 - 30
2.80 2.60 2.40 2.20 2.00 1.80 1.60 1.40 1.20 1.00 0.80 0.60 0.40 0.20 0.00
0.00 0.50 1.00 1.50 2.00 2.50 3.00 3.50 4.00 4.50 5.00 5.50
510
511
512
513
517
518

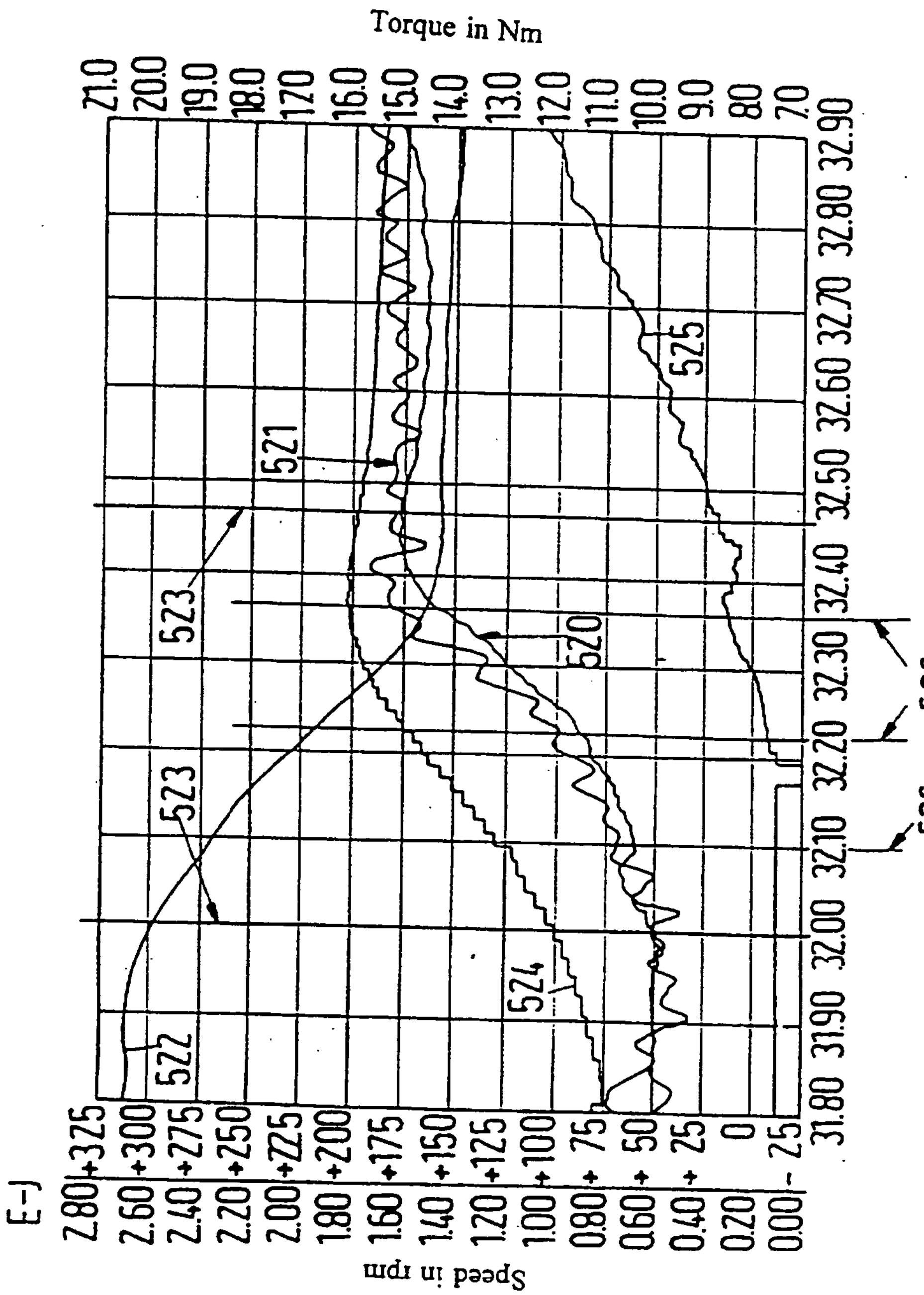

Torque in Nm
21.0
20.0
19.0
18.0
17.0
16.0
15.0
14.0
13.0
12.0
11.0
10.0
9.0
8.0
7.0
Speed in rpm
E-J
2.80 +325
2.60 +300
2.40 +275
2.20 +250
2.00 +225
1.80 +200
1.60 +175
1.40 +150
1.20 +125
1.00 +100
0.80 + 75
0.60 + 50
0.40 + 25
0.20 0
0.00 - 25
31.80
31.90
32.00
32.10
32.20
32.30
32.40
32.50
32.60
32.70
32.80
32.90
520
521
522
523
523
524
525
526
526

Fig.27

Fig. 28

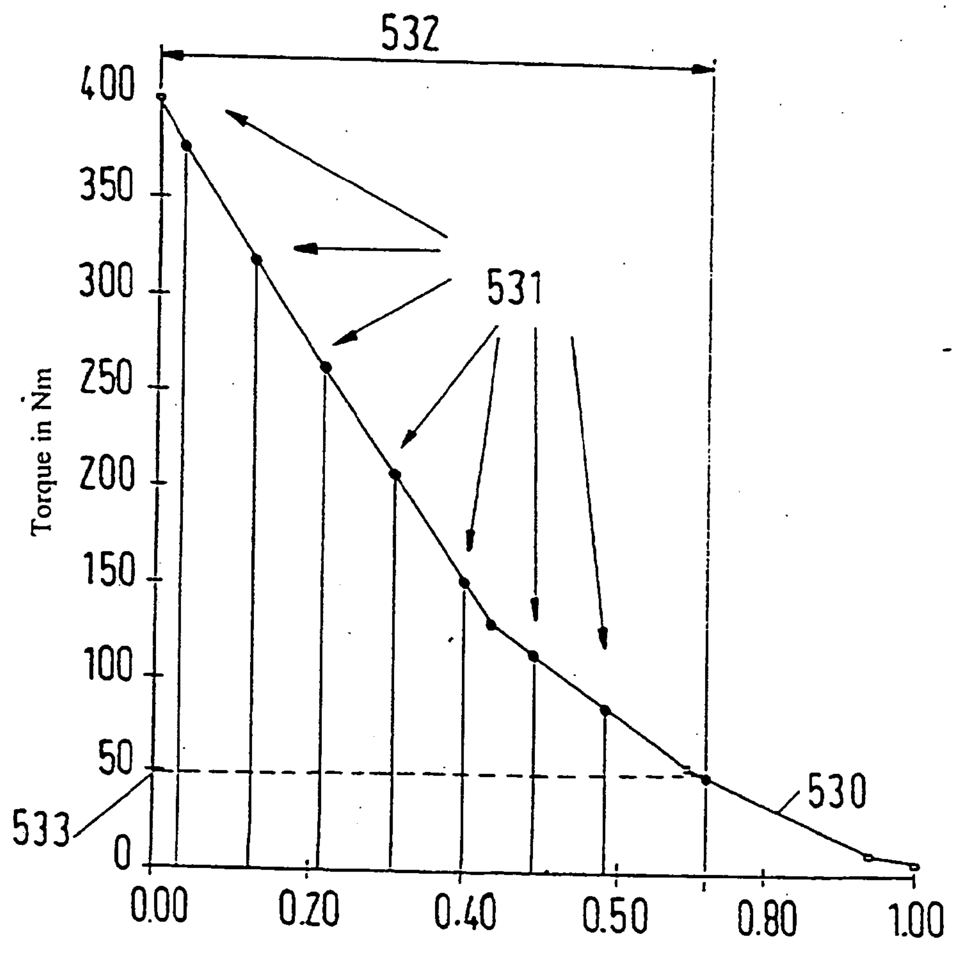

532
531
530
533
Torque in Nm
400
350
300
250
200
150
100
50
0
0.00
0.20
0.40
0.50
0.80
1.00

Fig.29a

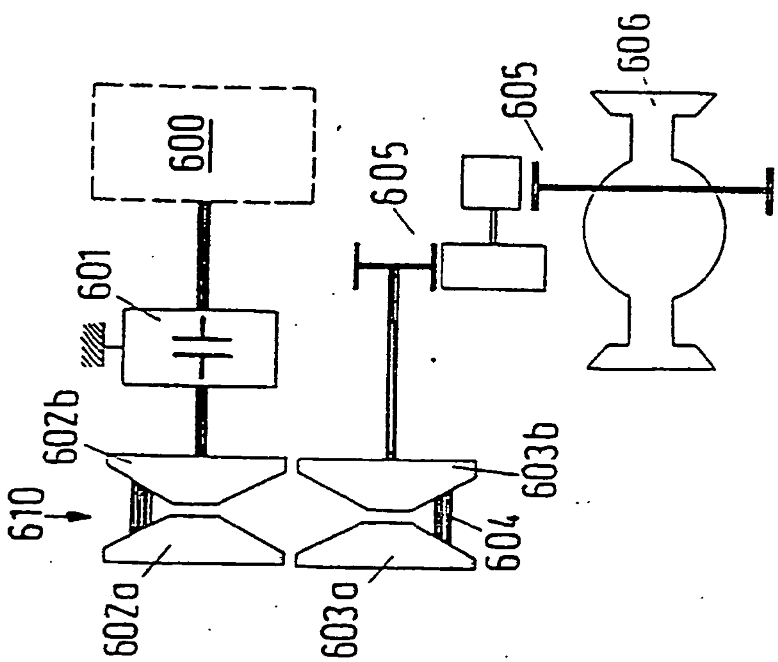
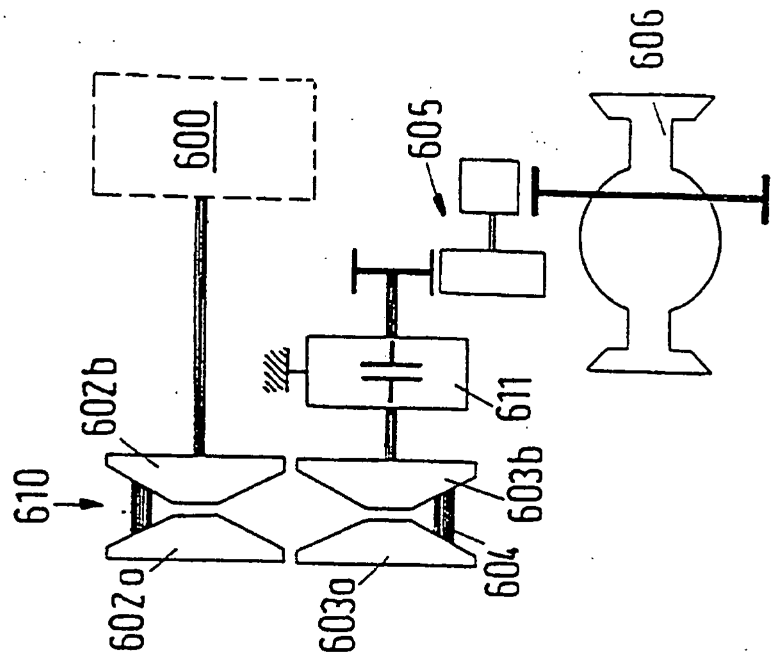


Fig.29b



Control Method For A Torque Transfer System And Torque Transfer System For Carrying Out The Control Method

The invention relates to a method for controlling a torque transfer system, a torque transfer system for carrying out the control method and a monitoring method for torque transfer systems.

It is known from the vehicle industry when changing the transmission ratio or a gear between a drive machine and a gear box unit to support or alternate the required clutch processes with a control or regulating algorithm. This makes the operation of the engine unit or gear box unit easier and the clutch method can be carried out in an energy saving manner with careful treatment of the materials. Furthermore the control of a torque transfer system which is mounted at the output side of an automatic gearbox can be helpful in undertaking or guaranteeing adjustment processes and protective functions in the case of for example cone pulley belt contact gearboxes.

From WO94/04852 a control method is known for torque transfer systems in conjunction with an automatic gearbox. The torque transfer system has a load distribution with a torque converter which is mounted parallel with a friction clutch. With this method a drive torque supplied by an engine unit is broken up into a hydraulic part which is to be transferred by the converter and a mechanical part which is to be transferred by the friction clutch, such as a lock-up clutch. A central control unit or computer unit determines or calculates in dependence on the relevant operating state of the system, the torque to be transferred each time by the friction clutch. The remaining torque to be transferred by the hydraulic torque

converter is produced from the difference between the adjoining torque and the torque transferred by the friction clutch and corresponds directly to a slip between the drive and output part of the torque transfer system.

This control method can only be used in conjunction with an automatic gearbox and a lock-up clutch. The acceptance of automatic gearboxes is however only slight in many areas of use. Furthermore a lock-up clutch of this kind is cost-intensive and requires a lot of space. The object of the invention is to provide a control method which can be used universally, which has high control quality with clearly improved load change behaviour for torque transfer systems.

In addition there should be cost advantages compared to conventional torque transfer systems. Furthermore a torque transfer system is to be provided for carrying out a control method of this kind.

According to the invention, there is provided a control method for a torque transfer system with or without load distribution wherein the clutch torque which can be transferred from a drive side to an output side of the torque transfer system is used as the control value and this control value is controlled by means of a setting member which is provided with a setting value which is functionally dependent on the transferable clutch torque, so that the transferable clutch torque always lies within a pre-determinable tolerance band about the slip limit wherein this slip limit is then accurately reached when the action of a torque arising on the drive side exceeds the clutch torque transferable by the torque-transferring parts.

The clutch torque which can be transferred from a drive side to an output side of a torque transfer system with or without load distribution can be used as the control value wherein this control value is calculated and/or determined in dependence on a drive torque.

The concept of torque matching is embodied here. The basic idea of a method of this kind lies in controlling the setting member mainly so that the clutch torque which can be transferred by the torque-transferring parts lies mainly just above or just below the drive torque which arises on the drive side of the torque transfer system.

A torque transfer system must generally be designed for two to three times the maximum drive torque of a drive machine, such as an engine. The drive torque which is typical for the operation lies however at a fraction of the maximum drive torque. The torque matching makes it possible to produce only the force locking engagement actually required between the torque-transferring parts instead of a quasi permanent high over pressure.

A further advantage lies in the use of a control method. In contrast to a regulation the feedback of condition values of the torque transfer system is not absolutely necessary. It only serves for a possible increase in the control quality but is however not required in order to produce the function of the torque transfer system. The task of a torque transfer system of this kind is the transfer of torque. It is therefore expedient to use the transferable clutch torque as the control value.

An advantageous design of the invention is characterised in that in the case of a method for controlling a torque transfer system with or without load distribution which

controls the torque transferable from a drive to an output side of the torque transfer system, there is a sensor system for detecting measured values and a central control or computer unit connected therewith wherein the torque transferable by the torque transfer system is controlled so that the transferable torque is calculated as a function of a drive torque, adapted and controlled and deviations from the ideal state are compensated long term through corrections.

Furthermore it can be advantageous if a method is used which serves to control a torque transfer system, more particularly for motor vehicles, wherein the torque transfer system is connected in the force flow after a drive machine and in the force flow in front of or behind a translation-variable device, such as a gearbox, and controls the torque transferable from a drive side to an output side of the torque transfer system, includes a control or computer unit which is in signal connection with sensors and/or other electronic units wherein the torque transferable by the torque transfer system is calculated as a function of a drive torque and adaptively controlled and deviations from the ideal state are compensated long term through corrections.

According to another design, the control value can be controlled by means of a setting member supplied with a setting value which is functionally dependent on the transferable clutch torque so that the transferable clutch torque always lies within a pre-determined tolerance band about a slip limit wherein this slip limit is then reached if the action of a torque arising on the drive side exceeds the clutch torque which can be transferred by the torque-transferring parts.

More particularly, the method according to this design can be carried out so that the torque transferable by a torque transfer system, such as a friction clutch and/or hydrodynamic torque converter with or without converter lock-up clutch and/or starting clutch for automatic gearboxes and/or turning set clutch and/or torque transfer system connected in front or behind an infinitely adjustable gearbox, such as a cone pulley belt contact gearbox can be controlled as a function of a drive torque so that in the case of systems with load distribution, such as hydrodynamic torque converter with converter lock-up clutch, the torque transferable by the clutch is determined according to the torque equation

$M_{KSoll} = K_{ME} * M_{AN}$ and

$M_{Hydro} = (1-K_{ME}) * M_{AN}$

wherein these two equations apply for $K_{ME} \leq 1$ and

$M_{KSoll} = K_{ME} * M_{AN}$ and

$M_{Hydro} = 0$

applies for $K_{ME} \rangle 1$ with

K_{ME} = torque division factor

M_{KSoll} = clutch ideal torque

M_{AN} = adjoining torque

M_{Hydro} = torque transferable by the hydrodynamic torque converter

and a torque difference between the torque M_{AN} adjoining the torque transfer system from the drive assembly and the torque M_{KSoll} transferable by the clutch is transferred through the hydrodynamic torque converter wherein a minimum slip is independently set between the drive and output of the torque transfer system in dependence on the torque division factor K_{ME} and deviations from the ideal state are adaptively detected, processed and compensated long term.

A further variation of the method according to the invention proposes that the torque transferable by the torque transfer system is controlled as a function of a drive torque so that in the case of systems without load distribution, such as friction clutch and/or starting clutch and/or turning set clutch and/or torque transfer system of an automatic gearbox or an infinitely adjustable gearbox, such as a cone pulley belt contact gearbox, the torque transferable by the friction clutch or starting clutch

$$M_{KSoll} = K_{ME} * M_{AN}$$

is detected and a definite over pressing of the torque-transferring parts is carried out for $K_{ME} \geq 1$.

Furthermore, it can be advantageous if the torque transferable by a torque transfer system is controlled as a function of a drive torque so that in the case of systems without load distribution, such as friction clutch and/or starting clutch and/or torque transfer system of an automatic gearbox and/or of an infinitely adjustable cone pulley belt contact gearbox, the torque transferable by the torque transfer system

$$M_{KSoll} = K_{ME} * M_{AN} + M_{Sicher}$$

is detected and for $K_{ME} < 1$ a fictitious load distribution copies through a supporting control loop the behaviour of a parallel-connected torque transfer system, such as hydrodynamic torque converter and a proportion of the transferable torque is controlled through the torque control and the remaining torque is subsequently controlled in dependence on slip through a safety torque M_{Sicher}.

Furthermore it can be advantageous if the safety torque M_{Sicher} is adjusted in dependence on each operating point.

Similarly it can be advantageous if the safety torque M_{Sicher} is detected and/or controlled in functional dependence on the slip n or the throttle valve position d according to

$$M_{Sicher} = S(n, d).$$

Similarly it can be expedient if the safety torque M_{Sicher} is detected and/or controlled according to

$$M_{Sicher} = Const. * n.$$

Furthermore it can be advantageous if the torque division factor K_{ME} is constant over the entire operating area of the drive train.

Similarly it can be advantageous if the torque division factor K_{ME} assumes an individual value detected from each relevant operating point and/or assumes at least in a partial area of the operating area a relevant constant value each time wherein the value set in different partial areas can be different.

It is thereby advantageously possible to divide the entire operating area into partial areas wherein in each relevant partial area the K_{ME} value is kept constant and the constant K_{ME} value can vary from operating area to operating area.

Furthermore, it can be advantageous if the value of the torque division factor K_{ME} is in functional connection dependent on the drive speed and/or the vehicle speed.

According to the idea of the invention it can be advantageous if the value of the torque division factor K_{ME} is solely dependent on the speed of the drive assembly.

Similarly it may be advantageous if the value of the torque division factor at least in a partial area of the entire operating area is dependent both on the speed and on the torque of the drive assembly.

Furthermore it may be advantageous if the value of the torque division factor K_{ME} is dependent on both the output speed and on the torque of the drive assembly.

Furthermore it may be advantageous if a certain ideal clutch torque is transferred by the torque transfer system substantially at each point in time. It can thereby be expedient if the transferable clutch torque matches the arising torque.

This design has the advantage that the contact pressure of the torque transfer system need not be kept permanently at the highest value. According to the prior art a torque transfer system, such as a clutch is biased with a multiple of the nominal engine torque.

With the automated torque transfer system the copying of the transferable torque has the result that the regulating unit or actor not only controls opening and closing processes during switching and starting but that the regulating unit sets the transferable torque in each operating point to a value which corresponds at least substantially to the ideal value.

So that the regulating unit or actor need not be constantly active when matching, it may be expedient if the transferable torque of the torque transfer system is controlled with an over pressure and the over pressure lies within a small scatter band in relation to the ideal

value.

It can be expedient if the over pressure M is dependent on the operating point.

More particularly it can be advantageous if the operating area is divided into partial areas and the contact pressure and/or the maximum over pressure is fixed for each partial area.

In a further embodiment of the invention it may be advantageous if the contact pressure and/or over pressure and/or the transferable clutch torque is controlled variable in time.

Similarly according to the inventive idea it may be advantageous if the transferable clutch torque which is to be set does not under step a minimum value M_{Min}. The minimum torque can depend on the operating point and/or on the momentary operating area and/or on the time.

Furthermore the torque matching can be carried out by means of a combination of a time variable matching with a minimum value which is specific to the operating point.

According to the inventive idea it can be advantageous if an operating point or a relevant operating state of a torque transfer system and/or of an internal combustion engine is determined from the condition values detected or calculated from the measured signals, such as in dependence on the engine speed and the throttle valve angle, in dependence on the engine speed and the fuel throughput, in dependence on the engine speed and the inlet manifold underpressure, in dependence on the engine speed and the injection time or in dependence on the

temperature and/or friction value and/or slip and/or the load lever and/or the load lever gradient.

Advantageously with a torque transfer system with an internal combustion engine mounted on the drive side the drive torque of the internal combustion engine can be determined from at least one of the condition values of the operating point, such as the engine speed, throttle valve angle, fuel throughput, inlet manifold underpressure, injection time or temperature.

A further different variation of the method proposes that the torque $M_{AN} * K_{ME}$ adjoining the torque transfer system on the drive side is influenced and/or altered with a dependence taking into account the dynamics of the system wherein the dynamics of the system can be cooled through the dynamic behaviour as a result of the mass inertia torque and/or free angles and/or damping elements.

It can be advantageous if means are provided which deliberately restrict or influence the dynamics of the system.

Similarly it can be advantageous if the dynamics of the system are brought about to influence $M_{AN} * K_{ME}$ in a form of the gradient restriction.

The gradient restriction can be effected as a limitation of a permissible increment.

Furthermore it can be advantageous if the gradient restriction is effected so that the time change and/or the time changed rise of a signal is compared with the maximum permitted slope or slope function and where the maximum permissible increment is exceeded the signal is replaced

by a replacement signal which is incremented with a previously defined slope.

Furthermore it can be advantageous if the control or restriction of the dynamics of the system is designed according to the principle of a time dynamic and/or variable filter wherein the characteristic time constants and/or amplifications are time variable and/or dependent on the operating point.

Advantageously the dynamics of the system are taken into account and/or processed with a PT_1-filter.

It can likewise be advantageous if the dynamics of the system are marked by a maximum restriction wherein on exceeding a certain boundary value the ideal value is represented by the boundary value and correspondingly the ideal value does not exceed a maximum value which is represented by the boundary value.

Furthermore it can be advantageous if at least two means for controlling the system, such as a gradient restriction and a filter stage, are connected in series.

It can likewise be advantageous if at least two means for influencing the dynamics of the system such as a gradient restriction and a filter are connected in parallel.

More particularly it is advantageous if the dynamics of the internal combustion engine and the dynamics of the secondary consumer which cause a load distribution are taken into account when determining the drive torque M_{AN}. In these cases it is more particularly advantageous if the mass inertia torque of the relevant flywheel masses and/or elements are used to take into account the dynamics of the

internal combustion engine.

It can likewise be advantageous if the injection behaviour of the internal combustion engine is used for and/or forms the basis of the consideration of the dynamics of the internal combustion engine.

Likewise within the scope of the control method according to the invention deviations from the ideal state can be compensated long term by considering the secondary consumer and/or the correction and/or the compensation of breakdowns and/or breakdown sources.

It can be advantageous if the torque adjoining the torque transfer system on the input side is detected and/or calculated as a difference between the engine torque M_{mot} and the sum of the torques of the secondary consumer taken up or branched off. As secondary consumers can be considered for example the climate control and/or the dynamo and/or servo pumps and/or steering aid pumps.

According to the idea of the invention it can be advantageous if system condition values, such as the engine speed and the throttle valve angle, the engine speed and the fuel throughput, the engine speed and the inlet manifold underpressure, the engine speed and the injection time, the engine speed and the load lever are used to determine the value of the engine torque M_{mot}.

Furthermore it can be advantageous if by means of system condition values the engine torque M_{mot} is detected from an engine characteristic field. Correspondingly it can be advantageous if system condition values are used to determine the engine torque M_{mot} and the engine torque is determined through the solution of at least one equation

or an equation system. The solution of the equation or the equation system can be carried out numerically and/or can be detected from the characteristic field data.

Furthermore it can be advantageous if the torque take-up or the load distribution of the secondary consumer is determined from measured values, such as voltage and/or current measured values of the dynamo and/or switch-on signals of the relevant secondary consumer and/or other signals indicating the operating state of the secondary consumer.

Furthermore it can be advantageous if the torque take-up of the secondary consumer is determined by means of measured values from the characteristic fields of the relevant secondary consumer. Likewise the torque take-up of the secondary consumer can be determined by solving at least one equation or an equation system.

According to the idea of the invention it can be expedient if the corrected transferable clutch torque can be determined according to the torque equation

$$M_{KSoll} = K_{ME} * (M_{AN} - M_{Korr}) + M_{Sicher}$$

and the correction torque M_{Korr} is produced from a correction value which is dependent on the sum of the moments taken up or branched off from the secondary assemblies.

Furthermore it can be advantageous if a correction is carried out of breakdowns which have an effect on measurable system input values.

More particularly it can be advantageous for the method according to the invention if measurable breakdown factors are detected and/or identified and at least partially

compensated and/or corrected through a parameter adaptation and/or a system adaptation. Furthermore it can be advantageous if measurable system input values are used in order to identify breakdown values and/or to correct and/or to compensate at least partially these values through parameter adaptation and/or system adaptation.

In order to identify a breakdown factor and/or to correct same by means of a parameter adaptation and/or system adaptation and/or to compensate same at least partially it is possible to use system input values such as for example temperatures, speeds, friction value and/or slip as the values.

More particularly it can be advantageous for the method if a compensation and/or correction of measurable breakdown factors is carried out through adaptation of the engine characteristic field.

In these cases it may be the case that a breakdown value is observed or registered which originally need not be in connection with the engine characteristic field but a correction of this breakdown value can be advantageous through an adaptation of the engine characteristic field. In this case the cause of the breakdown value is not corrected or compensated.

It can be advantageous if from a comparison between the clutch ideal torque and clutch actual torque a correction characteristic line field is produced and for each operating point a correction value is detected or can be detected which is linked additively and/or multiplicatively with the value of the engine torque from the engine characteristic field.

Furthermore it can be particularly expedient if in the light of a deviation detected in an operating point ideal value and actual value analysis and/or measures are introduced in order to calculate and/or fix deviations and/or correction values in other operating points of the entire operating area.

Furthermore it can be advantageous if in the light of a deviation detected in an operating point analysis and/or measures are introduced in order to calculate or fix deviations and/or correction values in other operating points of a restricted operating area. In relation to the method it can be advantageous if the restricted operating areas are fixed in dependence on the characteristic field.

Advantageously an embodiment of the invention can be characterised in that the analyses and/or measures for determining and/or calculating deviations and correction values in the additional operating points take into account the entire or a restricted operating area.

Furthermore it can be advantageous if the analyses and/or measures for calculating deviations and/or correction values in the further operating points only detect partial areas around the actual operating point. More particularly it can be advantageous if the analyses and/or measures for determining and/or calculating deviations and/or correction values are carried out in the further operating points so that weighting factors evaluate or weigh up different areas of the entire operating area differently.

Furthermore it can be advantageous if the weighting factors are selected and/or calculated as a function of the operating point. It can likewise be advantageous if

the weighting factors can be dependent on the type of breakdown value and/or on the cause of the breakdown.

Furthermore it can be advantageous in particular if after determining the correction value and/or after evaluating the correction characteristic field a time behaviour is imprinted on the correction value. This time behaviour may take into account for example the dynamic behaviour of the system.

Furthermore it can be advantageous if the time behaviour is determined through a beat frequency, a scanning of the correction value, and/or the time behaviour is determined through at least a digital and/or analogue filter.

More particularly it can be advantageous in the case of a design according to the invention if for different breakdown values and/or different breakdown sources the time behaviour is varied, ie in the event of using a relevant filter the parameters of the filter are set in dependence on the type and manner of the breakdown source. The time constants and amplifications of the filter are thus adapted to the relevant breakdown sources in order to guarantee the optimum adaptation possible.

It can be advantageous if the time behaviour is selected in dependence on the value of the corrections. More particularly it can be advantageous if the drive torque is adapted with an adaptation method with greater or smaller time constants than the time constants of the adaptation method of the clutch torque. It is advantageous if the time constant lies in an area of one second to 500 seconds, but preferably in an area from 10 seconds to 60 seconds and more particularly preferably in an area from 20 seconds to 40 seconds.

In a further embodiment it can be expedient if the time constant is dependent on the operating point and/or that the time constant is selected and/or determined differently in different operating areas. Furthermore it can be advantageous if a compensation and/or correction of measurable breakdown values is carried out through adaptation of the inverse transfer function of the transfer unit with setting member.

A further advantageous variation of the method shows that indirectly measurable breakdown values, such as in particular the ageing and/or the mean variation of individual component parts of the torque transfer system are detected in that some characteristic values of the torque transfer system are monitored and in dependence on this monitoring the parameters actually disturbed are detected and corrected and/or virtual breakdown sources which can be switched on in the form of program modules are used in order to correct and/or compensate the influence of the breakdown values.

It can furthermore be advantageous if breakdowns from non-measurable influence values, the mean variation of individual component parts and/or the ageing are detected and/or compensated through deviations from condition values of the system. Furthermore it can be advantageous if breakdowns, such as the mean variation or ageing or other non-measurable influence values, are not detected from measurable input values but are only recognised by observing system reactions.

It can likewise be advantageous if the deviations from system condition values or condition values and/or observations of system reactions are measured directly

and/or calculated from other measured values in a method model.

It can likewise be advantageous to carry out the detection of deviations from calculated method models by means of reference characteristic fields and/or clear reference characteristic values of the system.

Another advantageous further development of the invention proposes that for the correction and/or for the compensation of a detected breakdown from non-measurable input values a breakdown source is localised and/or a breakdown source is fixed and the deviations at these breakdown sources are corrected and/or compensated. Furthermore it can be expedient if for the correction and/or for the compensation of a detected breakdown a fictional breakdown source is fixed which need not originally be responsible for the breakdown at which the detected deviation is corrected.

Advantageously the fixed breakdown source can be a function block actually existing and/or the fixed breakdown source is a virtual breakdown model whilst preserving the correcting action.

According to a further development of the invention the time path of the clutch actual torque is monitored and analysed to see whether statements on the type of error and/or the detection of the breakdown source and/or the localisation of the breakdown source can be provided.

Furthermore it can be advantageous if the adaptive correction of the breakdown value is permanently carried out.

A further advantageous design proposes that the adaptive correction of the breakdown values is only carried out in certain operating points and/or certain operating areas and/or time areas.

Furthermore it can be advantageous if the adaptation can be active if the control is inactive. Inactive here means that the control does not indicate or cause or carry out any activity of the setting member since for example an operating area is selected or actually exists in which a torque matching is not carried out but a stationery value is set. In this operating area an adaptation of the parameter can be carried out without carrying out an active control.

Furthermore it can be advantageous if the adaptation is not carried out in special operating areas, more particularly in the event of severe acceleration.

It can be expedient if in the operating areas of the inactive adaptation the correction values of the setting values are used which were detected in the previously determined operating areas of the active adaptation. For this procedure it may furthermore be expedient if the previously detected values for an adaptation are stored in an intermediate memory and can be called up in situations of a de-activated adaptation.

For a further embodiment of the invention it may be expedient if in the operating areas of the inactive adaptation the correction values of the breakdown values are applied which can be extrapolated from correction values in previously detected operating areas with active adaptation.

According to a further method according to the invention it can be expedient if virtual breakdown models and/or virtual breakdown values are adapted for the area of the engine torque and/or for the area of the net engine torque after taking into account the secondary consumer, and/or for the clutch ideal torque.

Furthermore it can be advantageous if the inverse transfer function of the transfer unit with setting member is used and/or applied as virtual breakdown source.

It can furthermore be expedient if the engine characteristic field is used as the virtual breakdown source.

More particularly it is advantageous if virtual breakdown sources are used to define breakdown values whose original causes cannot be localised, such as eg mean variation in the area of manufacturing tolerances of the individual component elements.

A further idea of the invention relates to a control method for a torque transfer system with or without load distribution wherein the clutch torque transferable from a drive to an output side of the torque transfer system is used as the control value and this control value is controlled by means of a setting member which is provided with a setting value which is functionally dependent on the transferable clutch torque, so that the transferable clutch torque always lies within a predetermined tolerance band around the slip limit wherein this slip limit is then accurately reached if the action of a torque arising on the drive side exceeds the clutch torque which can be transferred by the torque-transferring parts.

Furthermore it can be advantageous if a value is set on the setting member as a setting value which corresponds to the transferable clutch torque between the torque-transferred parts of the torque transfer system.

A further expedient development of the invention proposes that the setting value is determined in dependence on a transferable clutch torque and that to calculate this transferable clutch torque a difference is formed from the drive torque value and a correction value wherein this correction value is increased or reduced in dependence on at least one condition value of the torque transfer system.

Furthermore it can be expedient if the correction value is determined in dependence on a differential speed between a drive and an output speed, designated slip speed, wherein this correction value is increased so long as the slip speed lies below a predetermined slip boundary value and the correction value is reduced so long as the slip speed lies above this or another predeterminable slip boundary value.

Furthermore it can be advantageous if the correction value is increased incrementally so long as the slip speed lies below a slip boundary value and the correction value is reduced step wise so long as the slip speed lies above the one or another slip boundary value wherein between the relevant stages stopping phases of adjustable length exist within which the correction value is kept constant at the value set at the beginning of each stopping phase.

Furthermore it can be advantageous if the times in which the drive speed exceeds the output speed by a defined slip speed are recognised as the slip phase and at the end of

each slip phase the correction value is set again to a definite value.

An expedient design of the invention proposes that the times in which the drive speed exceeds the output speed by a definite slip speed are recognised as slip phases, and that the relevant correction value at which the slip speed assumes its maximum value is stored in an intermediate memory and at the end of each slip phase the actual correction value is again replaced by the stored correction value.

It can likewise be advantageous if the correction value at the end of each slip phase is kept constant at its relevant value for a fixable time duration. According to another design of the invention it can be advantageous if the setting member is provided with a preset value in dependence on a characteristic field or a characteristic line which includes the area of all possible transferable clutch moments or has at least one partial area within which only one preset value is allocated for the setting member to all transferable clutch moments.

It can furthermore be advantageous that to calculate the transferable clutch torque a difference is formed from a drive torque value and the correction value and that this difference is increased by a torque value which is dependent on slip.

According to a further design of the invention it may be favourable if the rise in the actual clutch torque is restricted in the form of a gradient restriction in that the relevant actual value of the transferable clutch torque is compared with a comparison torque value which consists of a previously detected transferable clutch

torque value and an additive fixable restriction value and that in dependence on this comparison the relevant smaller torque value is given to the setting member as the new preset value.

It can be advantageous in particular if several condition values, such as eg the engine speed, throttle valve angle and/or suction intake pressure are detected from an internal combustion engine mounted on the drive side of the torque transfer system and that from these condition values the drive torque of the combustion engine is detected by means of stored characteristic lines or characteristic line fields. Furthermore the invention proposes that some load forks lying between the drive and torque transfer system are monitored at least partially or at least temporarily and the measured values resulting therefrom are used to calculate the drive torque actually arising on the drive side of the torque transfer system.

It can be advantageous if a part of the drive torque corresponding to a proportion factor is used each time to calculate the transferable clutch torque and that this proportion factor is determined each time from the stored characteristic line fields or characteristic lines.

Furthermore it can be expedient if with torque transfer systems without load distribution a load distribution is imitated through a secondary control program.

According to the inventive idea it can be advantageous if measurable breakdown values, such as in particular temperatures and/or speeds are detected and are compensated at least partially through a parameter adaptation and/or through a system adaptation.

An expedient further development proposes that indirectly measurable breakdown values of the control method, such as in particular ageing and/or mean variation of individual component parts of the torque transfer system are detected in that some condition values of the torque transfer system are monitored and in dependence on this monitoring the parameters actually disturbed are recognised and corrected and/or virtual breakdown sources which can be switched on in the form of program modules are used in order to correct and/or compensate the influence of the breakdown values.

Furthermore it can be advantageous if a first engagement of the clutch is only possible after checking a user authorisation.

It can likewise be advantageous if a display, such as a user display, is controlled in dependence on the status of the control method so that a switching command is given for the user. This switching command can be carried out through the display in an optical or acoustic manner.

Further it can be advantageous if stationary phases, more particularly of a vehicle, are recognised by monitoring significant operating values, such as accelerator pedal and/or gear stick position and/or tacho speed and on exceeding a defined time period the drive unit is stopped and started again when necessary.

Furthermore it can be advantageous if operating phases of the torque transfer system with minimum or without load testing are recognised as freewheel phases and within these freewheel phases the clutch is opened and at the end of the freewheel phase the clutch is closed again. The end of the freewheel phase can be produced or recognised

for example through a detected change in the load lever position and/or load lever gradient.

According to a further design of the invention in order to assist an anti-blocking system the control method is used so that during the response of the ABS system the clutch is completely disengaged.

Furthermore it can be advantageous if the setting member is controlled in certain operating areas after presetting the anti-slip control.

The invention not only relates to the method described above for controlling a torque transfer system, but also in particular relates to a torque transfer system for transferring torque from a drive to an output side wherein an internal combustion engine such as a motor is mounted on the drive side and a gearbox is mounted on the output side and the torque transfer system has a clutch, setting member and a control device.

Furthermore the invention relates to a torque transfer system which can be controlled by means of the method described above and serves to transfer torque from a drive to an output side wherein the torque transfer system is connected in front or behind on the output side in the force flow of a drive unit, such as an internal combustion engine and in the force flow of a translation-variable device, such as a gearbox, and the torque transfer system has or includes a clutch and/or a torque converter with lock-up clutch and/or a starting clutch and/or a turning set clutch and/or a safety clutch restricting the transferable torque, a setting member and a control device.

More particularly it can be advantageous according to the idea according to the invention if the clutch is a self adjusting or self setting clutch.

Similarly it can be advantageous if the clutch automatically compensates the wear for example on the friction linings.

According to the inventive idea in the case of an embodiment of the invention it may be advantageous if to transfer the torque from a drive to an output side the torque transfer system has a clutch, setting member and a control unit wherein the clutch is in active connection with the setting member through a hydraulic pipe which has a clutch receiver cylinder and the setting member is controlled by the control apparatus.

A further advantage lies in the use of a setting member which has an electric motor which acts through an eccentric on a hydraulic transmitter cylinder which is attached to the hydraulic pipe which is connected to the clutch, and that a clutch path sensor is mounted in the housing of the setting member.

With a view to a space-saving flexible solution for the arrangement of the device according to the invention it is advantageous if the electric motor, eccentric, transmitter cylinder, clutch path sensor and the control and load electronics required are mounted inside a housing of the setting member.

It can likewise be of advantage if the axes of the electric motor and of the transmitter cylinder are mounted to run parallel to each other. It is particularly advantageous if the axes of the electric motor and of the

transmitter cylinder are mounted to run parallel to each other in two different planes and are in active connection through the eccentric.

It can furthermore be of advantage if the axis of the electric motor runs parallel to a plane which is formed substantially by the plate of the control and load electronics.

According to a further development of the torque transfer system according to the invention the functional method of the transfer system can be optimised by mounting a spring centrally with the axis of the transmitter cylinder in the housing of the setting member.

Furthermore it can be advantageous if a spring is mounted concentrically with the axis of the transmitter cylinder in the housing of the transmitter cylinder.

It can be advantageous for the functioning of the device according to the invention if a spring characteristic line of the spring is adapted so that the maximum force to be applied by the electric motor for engaging and disengaging the clutch is approximately the same size in the pull and push direction.

Furthermore, it can be advantageous if the spring characteristic line of the spring is designed so that the resulting force path of the forces acting on the clutch is made linear over the disengagement and engagement method of the clutch. According to a further development the power requirement and thus the size of the electric motor used is minimised. Forces required for the disengagement method of the clutch are decisive for the measurement of the electric motor to be used since for the disengagement

method a higher force action is required than for the engagement method of the clutch in that the force action of the spring supports the disengagement method and the electric motor can be made with a weaker output.

By using a spring within the transmitter cylinder system no additional space requirement is necessary through the spring.

Furthermore it can be of advantage if the electric motor acts with a motor output shaft through a worm on a segment wheel and a crank is attached to this segment wheel wherein the crank is in active connection through a piston rod with the piston of the transmitter cylinder so that pushing and pulling forces can be transferred.

It can likewise be advantageous if the worm forms with the segment wheel a self-locking gearbox.

The invention does not, however, only relate to the method described above for controlling a torque transfer system and to the torque transfer system itself, but also includes a monitoring method for a torque transfer system with a manual gearbox, wherein relevant gear lever positions and a drive torque of a drive unit on the drive side are detected with a sensor system and at least each one corresponding gear lever signal and at least one comparison signal are recorded and different possible characteristics of these signal paths, such as for example a difference are recognised and identified as the switching intention and a switching intention signal is then supplied to a clutch operating system on the output side.

Regarding the idea according to the invention it can be

advantageous if at least one gear lever signal path is evaluated to detect the gear and this data is used to identify a switching intention.

The monitoring method determines the gear engaged at that time wherein this data can be used to determine the comparison signal.

A method is hereby provided with which any possible switching intention of the user is recognised at high speed and with great reliability without a specific sensor being necessary.

A substantially automated torque transfer system requires early data on a possible switching intention in order to separate the clutch at the correct time.

It can be advantageous if a gear lever signal and a comparison signal are evaluated so that intersecting points of these signal paths are detected and then a switching intention signal is sent to clutch operating signal on the output side. If, in order to detect the switching intention only two signal paths are investigated or evaluated for intersecting points, there is no longer any need for expensive software or hardware.

According to the idea of the invention it can be advantageous if with the switching gearbox a selection path is differentiated between the switching lanes and a switching path within the switching lanes wherein in order to determine the relevant gear lever position the switching path and/or the selection path can be detected.

There is also no need for additional sensor systems to form the comparison signal since the single input value,

the drive torque, can as a rule already be determined. Since the comparison signal is formed from the filter signal wherein the filter signal is increased and/or reduced by a constant value and an offset signal, it is substantially ensured that the gear lever signal and the comparison signal only then intersect if there is actually any switching intention.

In an advantageous development the presence of a switching intention is detected when evaluating the two signal paths of the gear lever signal and comparison signal if an intersection point is detected wherein the switching intention is verified by means of a switching intention counter. Through the claimed switching intention counter it is ensured that between the recognition of the switching intention and the sending of the switching intention signal, there is a definite time period in which it is checked whether a switching method is actually introduced. The torque transfer system is thus effectively secured against a faulty release.

The gear lever signal is filtered in order to form a filter signal with an adjustable delay time.

It can be particularly advantageous if the gear lever signal can be processed to form the filter signal with a PT 1 - behaviour.

Furthermore it can be advantageous if the gear lever signal is monitored and a change in the switching path within a defined partial area of the gear lever path is evaluated within a fixable measuring period so that when a fixable switching path change threshold is understepped a switching intention is sent to devices on the output side.

The gear lever signal which is used to determine the switching intention which is passed on in turn, can be adapted by means of individually adjustable filters which are universally usable through parameters so that the most varied torque transfer systems can be monitored with the same method. It is advantageous if the measuring period is fixed so that it is always clearly greater than a half vibration period or vibration amplitude of the gear lever which is not operated during driving operation.

It can be expedient if the defined partial area of the gear lever path is outside of the gear lever path areas within which the non-operated gear lever moves in the driving operation.

In order to carry out the method according to the invention, it is as a rule necessary to average out the gear lever vibration periods, thus the length of the measuring periods can be fixed in dependence on a mean value formation of the gear lever vibration periods.

In a further development it can be detected whether the gear lever vibrates freely in the driving operation, or more particularly when applying a hand, has a different vibration behaviour and that the mean value formation to determine the length of the measuring periods is carried out in dependence on the results of this monitoring.

According to a further embodiment of the invention it can be advantageous if the direction of movement of the gear lever is detected and when reversing this direction of movement a control signal is sent to the switching intention counter and/or a switching intention signal which has been given is rescinded.

The direction of movement of the gear lever is thereby additionally observed and in the event of a reversal of this direction of movement a switching intention signal is rescinded which would otherwise be given as a result of the vibrations of the gear lever.

Furthermore it can be advantageous if the constant value to form the comparison signal is selected in dependence on the typical operating vibration amplitude of the non-operated gear lever of the torque transfer system.

It can likewise be advantageous if the delay time with which the filter signal is formed, is adapted to the vibration frequency of the gear lever which is not operated during driving.

According to the idea of the invention it can be particularly advantageous for a control method if the drive load is monitored and on exceeding a fixable drive load a control signal is passed on to the switching intention counter. It can thus be prevented that in the event of an increased torque adjoining the engine side the clutch is opened or closed undesirably. It can likewise be advantageous if the offset signal is used in dependence on the relevant throttle valve angle of a combustion engine used as the drive unit.

According to the inventive idea it is expedient if the switching or selection path of the gear lever is detected by a potentiometer. It can likewise be advantageous if the switching and/or selection path of the gear lever is detected by a potentiometer in such a way that the gear position can be recognised by means of the potentiometers.

The invention relates not only to the method described above for controlling a torque transfer system but also includes those processes for controlling a torque transfer system having a device for controlling the torque transfer system, the torque transfer system is mounted on the output side in the force flow of a drive unit and on the input and/or output side in the force flow of a translation-variable device, the translation-variable device is provided with contact means which transfer a torque from a first means to a second means wherein the first means is in active connection with a gearbox input shaft and the second means is in active connection with a gearbox output shaft, the contact means is in friction connection with the first and the second means through contact pressure or tensioning and the contact pressure or tensioning of the contact means is controlled in dependence on the operating point, characterised in that the torque transfer system is controlled matching the torque, with a transferable torque which is dimensioned at each operating point so that the contact means of the translation-variable device does not begin to slip. This means that the slip limit of the torque transfer system is controlled in each operating point so that the slip limit of the contact means is always greater and in the event of too much torque the torque transfer system always begins to slip before the contact means slips.

Furthermore it can be advantageous if the contact pressure and/or tensioning of the contact means at each operating point is determined in dependence on the ensuing engine torque and/or the load distribution regarding the secondary consumer and an additional safety tolerance and the transferable torque of the torque transfer system is controlled in dependence on the operating point and the torque transferable by the torque transfer system leads,

in the event of torque fluctuations, to a slipping of the torque transfer system before the slip limit of the contact means is reached.

It is more particularly expedient if the slip limit of the torque transfer system at each operating point is lower than or is controlled lower than the slip limit of the contact means of the translation-variable device.

It can furthermore be advantageous according to the idea of the invention if the torque transfer system with its slip limit dependent on the operating point isolates and/or dampens torque fluctuation and torque impacts on the drive side and/or on the output side and protects the contact means against slipping. Slipping of the contact means is protected in the described cases wherein the slipping of the contact means could lead to destruction of the contact means and thus to a breakdown of the gearbox.

According to the inventive idea it is expedient to control the contact pressure or tensioning of the contact means in dependence on the operating point and in addition to the ensuing torque to take into account a safe reserve which can be matched and/or adapted to this transferable torque through the control of the transferable torque of the torque transfer system. Adapting the safety torque can be carried out in this case in that the design of the safety reserve can be made lower than that compared to the prior art.

It can be particularly advantageous if the safety reserve of the contact pressure or tensioning turns out as low as possible as a result of the slip protection of the torque transfer systems.

It is particularly expedient if the torque transfer system in the event of torque peaks slips or slides briefly. It is thus possible to isolate or dampen or filter torque impacts on the drive or output sides which may occur in extreme driving situations and could damage or destroy contact means.

The invention relates not only to the method described above but also to a device, such as a translation-variable device, which is controlled by means of the above mentioned method wherein the translation-variable device can be an infinitely adjustable gearbox. More particularly it may be advantageous if the translation-variable device is an infinitely adjustable cone pulley belt contact gearbox. More particularly it can be advantageous if the torque transfer system which is part of the device is a friction clutch or a converter lock-up clutch or a turning set clutch or a safety clutch. The clutch can be a dry or wet type clutch. Furthermore it may be expedient if a setting member controlling the transferable torque is provided which is controlled electrically and/or hydraulically and/or mechanically and/or pneumatically or the control of the setting member is produced through a combination of these features.

The invention not only relates to the method described above but also to a device with at least one sensor for the detection of the engaged transmission or of the engaged gear of a gear box wherein a central computer unit processes the sensor signals and calculates the gearbox input speed. For this calculation it is necessary to take into account translations, such as differential translations.

It can be advantageous if the detected wheel speeds are

averaged and the gearbox input speed is determined or calculated from this mean signal by means of the translations in the drive train and by means of the gearbox transmission.

It is advantageous if one to four sensors are used for determining the wheel speed, more particularly it is advantageous if two or four sensors are used.

The device can be designed in a particularly advantageous manner if the sensors for detecting the wheel speeds are in signal connection with an anti-blocking system or are component parts of an anti-blocking system.

The invention is explained in further detail with reference to an embodiment from the vehicle industry.
In the drawings:

Figure 1a is a block circuit diagram with a torque transfer system with a load distribution;

Figure 1b is a block circuit diagram with a torque transfer system without load distribution, wherein a fictional load distribution is copied through a secondary control programme;

Figs 2a to 2e are diagrammatic illustrations of different physical properties of a torque transfer system as a function of the torque division factor K_{ME}; 2a: acoustics as a function of K_{ME}; 2b: thermal load as function of K_{ME}; 2c: pulling force as a function of K_{ME}; 2d: fuel consumption as a function of K_{ME}; 2e: load change behaviour as a function of K_{ME};

Figure 3 is a block circuit diagram or a signal diagram of a control method with adaptation;

Figure 4 is a block circuit diagram or a signal diagram of a control method with adaptation;

Figures 5a to 5c show the effect of breakdown values on the time development of the torque; a: additive interference through e.g. additional assemblies; b: multiplicative interferences; c: additive breakdown values;

Figure 6 is an engine torque correction characteristic field as a function of the engine torque and the speed;

Figure 6a is a diagrammatic illustration of a division of a characteristic field;

Figure 6b is a diagrammatic illustration of a division of a characteristic field;

Figure 7 is a block circuit diagram for the control method with adaptation;

Figure 8 is a block circuit diagram for a control method with adaptation;

Figure 9 is a block circuit diagram for a control method with adaptation;

Figure 10 is a principle illustration of a vehicle with a torque transfer system;

Fig 11a is a longitudinal sectional view through a setting member unit of a torque transfer system;

Fig 11b is a cross sectional view of the setting member unit at III;

Fig 12a is a longitudinal sectional view through a setting member unit of a torque transfer system;

Fig 12b is a cross sectional view of the setting member unit at IV;

Fig 13 is a force diagram for the setting member behaviour;

Figure 14 is a diagram for determining a clutch torque;

Figure 15 shows a characteristic line field for determining a setting member target;

Figures 15a to 15e each show a diagram of the setting member target as a function of the time;

Figure 16 is a circuit diagram of a manual gearbox;

Figure 17 is a signal diagram for detecting the switching intention;

Figure 18 is a signal diagram for forming a comparison signal;

Figure 19 is a further signal diagram for detecting the switching intention;

Figure 20 is a signal diagram for verifying the detection of the switching intention;

Figure 21 is a functional diagram of an electro-hydraulically controlled torque transfer system;

Figure 22 shows a characteristic line;

Figure 23 is a block circuit diagram;

Figure 24 shows a signal path as a function of the time;

Figure 25 shows a signal path as a function of the time;

Figure 26 shows a signal path as a function of the time;

Figure 27 shows a signal path as a function of the time;

Figure 28 shows a characteristic line with support position adaptations;

Fig 29a shows a gearbox with a torque transfer system arranged at the input side and

Fig 29b shows a gearbox with a torque transfer system arranged at the output side.

Figures 1a and 1b each show a diagrammatic illustration of part of a drive train of a vehicle where a drive torque is transferred by an engine 1 with a mass inertia moment 2 to a torque transfer system 3. The torque transferable by this torque transfer system 3 can be transferred for example to a component part, such as an input part, of a

gearbox connected in series and not explained in further detail.

Figure 1a shows a diagrammatic illustration of a torque transfer system with load branching or distribution wherein for example a Föttinger clutch or a hydrodynamic torque converter 3a is arranged in the force path and is connected in parallel with a converter lock-up clutch 3b. A control device controls the torque transfer device 3 so that in at least some operating conditions the ensuing torque is transferred in parallel either only by the hydrodynamic torque converter 3a, only by the Fottinger clutch or the converter lock-up clutch 3b or by both the two torque-transferring devices 3a, 3b.

In some operating areas a deliberate division of the transferable torque between the relevant parallel torque-transferring devices 3a, 3b may be desirable and can be carried out accordingly wherein the ratio of the relevant torque transferred by for example the converter lock-up clutch 3b and the hydrodynamic torque converter 3a can be adapted to the special requirements of the individual operating areas.

Figure 1b shows a diagrammatic illustration with a torque transfer system 3 without load distribution. A torque transfer system 3 of this kind without load distribution can be for example a clutch, such as a friction clutch and/or a turning set clutch and/or a starting clutch and/or a safety clutch. A secondary control programme thereby copies a fictitious distribution load and controls the torque transfer system accordingly.

The diagrammatic sketches or block circuit diagrams of Figures 1a and 1b of a drive train, partially illustrated,

with a torque transfer system 3 mounted in the force flow in the drive train with or without load distribution only represent examples of possible arrangements or designs of torque transfer systems.

Furthermore arrangements of torque transfer systems are also possible wherein the relevant torque transfer system can be mounted in the force flow before or after the or each component part determining the gearbox transmission. Thus for example a torque transfer system, such as a clutch, can be mounted in the force flow before or after the variator of an infinitely adjustable cone pulley belt contact gearbox.

An infinitely adjustable gearbox, such as an infinitely adjustable cone pulley belt contact gearbox can likewise be produced with a torque transfer system mounted on the drive side and/or output side.

The systems with load distribution according to Figure 1a, such as a hydrodynamic torque converter 3a with lock-up clutch 3b can likewise be controlled by means of a control method according to the invention so that the torque transferable by the individual parallel-connected transfer systems, such as torque converter 3a and/or lock-up clutch 3b, is controlled accordingly. As a rule the torque to be transferred by one of the two torque transfer systems arranged in parallel is controlled and the torque transferable by the torque transfer system connected in parallel therewith is set automatically.

In the case of torque transfer systems with more than two (N) parallel connected transfer systems, as a rule the relevant transferable moments must be controlled by (N-1) transfer systems and the transferable torque of the (N-th)

transfer system is then set automatically.

In the case of systems without load distribution, such as e.g. a friction clutch, the transferable torque can be controlled through a control loop which underlies the control, so that by means of the control a system is simulated with fictitious load distribution. The friction clutch 3c is controlled with this control e.g. to an ideal value which is lower than 100% of the transferable torque. The difference between the torque ideal value thus controlled and the 100% of the entire transferable torque is controlled by a means of a control through a slip-dependent safety torque 3d. It is thereby reached that the friction clutch on the one hand is not closed with a higher contact pressure than would be necessary according to the torque to be transferred and secondly, as a result of the slipping operating state damping of the torsion vibrations and torque peak values, such as torque impacts, in the drive train can be guaranteed.

In another operating state of the operating area of the torque transfer system it can be advantageous if the torque transfer system, such as clutch or friction clutch, is controlled with a low but well-defined excess contact pressure. In these operating areas, e.g. at high speeds, it is thereby possible to avoid increased slip and thus fuel consumption of the internal combustion engine.

With a contact pressure of about 110% of the mean ensuing torque it is possible with brief torque peak values for a deliberate slipping or sliding of the clutch to take place. Thus damping of the peak values can be achieved with a substantially closed clutch.

With only a slight excess contact pressure of the clutch

it is further possible for torque impacts with peak values to be damped or isolated through brief slipping or sliding of the clutch.

The parameter characterising the division of the torque between the parallel mounted torque-transferring systems of the torque transfer system 3, is the torque division factor K_{ME} which is defined through the ratio between the torque transferable by a clutch or another torque-transferring system, such as e.g. a converter lock-up clutch, and the entire torque transferable by the torque transfer system.

The torque division factor K_{ME} thus indicates the ratio in which the transferable torque, e.g. of a clutch 3b stands in relation to the overall transferable torque.

When a K_{ME} -value is less than 1 this means that the transferable torque is divided between the parallel connected systems 3a, 3b, and the torque transferred by the relevant individual systems 3a, 3b is less than the overall torque which results or is to be transferred.

When $K_{ME} = 1$, the transferable torque is only transferred by one of the parallel mounted systems 3a, 3b, more particularly by the clutch 3b. With temporary torque peaks with values which lie above the value of the transferable torque, it can result in slipping or sliding of the clutch or torque transfer systems. However in the operating area without torque peaks the entire torque is transferred by one system 3a, 3b.

With a K_{ME} -value greater than 1 the entire arising torque is likewise transferred by one system, but for example, the contact pressure of the clutch corresponds to a

transferable torque which is greater than the ensuing torque. It is thereby possible to filter off greater torque irregularities which lie above a threshold value and slight torque irregularities are not filtered.

A further advantage of a defined excess contact pressure as opposed to the completely closed clutch is the shorter reaction time of the system until, for example, the clutch is opened. The system need not open the clutch from the completely engaged position but only from the currently set position. However, a slightly slower actuator can be used with the same time length.

Figures 2a to 2e show the behaviour of physical properties of physical values of torque transfer systems as a function of the torque division factor K_{ME}, on the example of a hydrodynamic torque converter with converter lock-up clutch. The plus and minus signs on the ordinate axis refer to a more positive or more negative influence of the K_{ME} factor on the physical properties illustrated.

Figure 2a shows the acoustic properties of the drive train of a motor vehicle, wherein a curve path of a torque transfer system with damper and a path of a torque transfer system without damper are shown as a function of K_{ME}. Both curves for the torque transfer systems with and without damper run parallel as a function of K_{ME}. The torque transfer system with damper has a slightly improved quality regarding acoustics compared to the torque transfer system without damper. As a function of the K_{ME} value it is seen that for $K_{ME} = 0$ the acoustics assume their most favourable value. With increasing K_{ME} the acoustic properties drop monotonously until with high K_{ME} values the acoustic properties show a change to a K_{ME} independent constant path.

This behaviour of the acoustic properties in dependence on the torque division factor K_{ME} can be explained through the increased uncoupling of the drive train from the torque irregularities and torque peaks of the drive assembly as a result of an increase in slip as a function of a reduced K_{ME} value.

With decreasing slip in the torque transfer system and increasing K_{ME} the torque irregularities in the drive train are transferred more strongly and the damping action is reduced simultaneously until with a certain K_{ME} value the damping becomes minimum or no longer exists at all. A constant acoustic behaviour thus results as a function of a further rising K_{ME} value.

The K_{ME} value at which a constant acoustic behaviour arises as a function of the torque division factor is dependent on the relevant characteristic of the drive train. With characteristic systems, this value lies at about $K_{ME} = 2$. With this value the clutch of the torque transfer system is closed so far that substantially each torque fluctuation is transferred.

Figure 2b shows the thermal load of a hydrodynamic torque converter with converter lock-up clutch as a function of the K_{ME} value. By thermal load can be meant for example the energy input into the system as a result of friction or as a result of differential speeds of the component parts. More particularly, for example, the energy input in a torque converter or into the fluid of a torque converter can be especially considered. The energy input into the friction faces of a converter lock-up clutch and/or friction clutch can likewise be understood here.

The low value of the thermal load for $K_{ME}= 0$ rises with an increasing K_{ME} value. By thermal load of this system is meant inter alia the energy input as a result of speed differences. With an increasing K_{ME} the energy input decreases as a result of speed differences in the converter until at $K_{ME} = 1$ the converter lock-up clutch is closed and the speed differences = 0 and thus the thermal load assumes its most favourable value. For $K_{ME} \geq 1$ the thermal load is constant and equal to the value for $K_{ME} = 1$.

Figure 2c shows the change in the pulling force which decreases as a function of a rising K_{ME} value since with a low K_{ME} value the conversion area of a torque converter is better utilised and/or the low K_{ME} allows another more favourable operating point of the internal combustion engine to be achieved.

Figure 2d can show a consumption which becomes more favourable as the K_{ME} value rises. Through a reduced slip for example in the area of the hydrodynamic torque converter it is possible for the fuel consumption to be reduced through a clutch which becomes increasingly closed as the K_{ME} value rises.

Figure 2e shows the load change behaviour as a function of the K_{ME} value. The load change behaviour is shown to be best with $K_{ME} = 1$, i.e. with a clutch closed in this way its transferable torque corresponds exactly to the adjoining torque.

Figure 3 shows a diagrammatic illustration of a block circuit diagram of a control method. In this diagram the setting member and control path are shown in a cohesive block 4. The control method 5 and the adaptation 6

(system adaptation and/or parameter adaptation) can likewise each be shown in encroaching blocks.

The control path with setting member or transfer unit with setting member 31 and the interferences acting on this system are shown in block 4. The drive assembly 16, such as internal combustion engine or motor, releases an engine torque M_{mot} 33 in dependence on the input values 14, such as for example injection amount, load lever, speed of the drive assembly etc or the system characteristic values 32, such as temperature etc. This engine torque M_{mot} 33 is branched off in part through secondary consumers 34, such as dynamo, climate control, servo pumps, steering aid pumps, etc. These secondary consumers are taken into account in the block 35 by subtracting the forked torque 34a from the engine torque 33 to form a resulting net torque 36.

The dynamics of the engine 16 and/or drive train, such as e.g. as a result of the mass inertia torque of the flywheel is considered in block 37. The dynamics can take into account more particularly the inertia moments of the relevant component parts and the effect of these inertia moments on the net drive torque. The torque M_{dyn} 38 corrected with regard to the dynamics of the system is transferred through a transfer unit with setting member 31 and from there is passed as the clutch actual torque 48 to the gearbox or to the vehicle 39 connected to the output side.

The transfer unit 31 with setting member is influenced by the values 40, such as temperature, friction value of the friction linings, speeds, slip, etc. In addition the transfer unit 31 like the motor 16 can be disturbed and/or influenced through mean variation, ageing or interferences

from influence values which cannot be directly measured. This influence is shown through the block 41.

The adaptation 6 can be divided basically into three areas. On the one hand the secondary consumers or secondary assemblies 7 are taken into account and the adaptation strategies or adaptation processes in connection therewith are used for the adaptation of the breakdown values and breakdown influences. Such secondary consumers can be climate control, dynamo, steering aid pump, servo pump and further secondary consumers which cause a division or branching of the torque.

For the adaptation of the secondary consumers 7 signals and data 8 of the secondary consumers 7 are used in order to be able to determine and/or calculate the relevant status of the secondary consumers 7. The status indicates inter alia whether the relevant secondary consumer branches off a torque because it is switched on or switched off and if it is switched on, how great the forked torque is at each relevant time point.

In Figure 3 it is clear that the system adaptation next to the secondary consumer adaptation 7 differs between and first and a second adaptation loop 9, 11. In the first adaptation loop 9 the influences of measurable breakdown values 10 are considered. In the second adaptation loop 11 influences of only indirectly measurable breakdown values or mean variations in the light of the directly measurable deviations and system condition values 12 are detected.

A correction and/or compensation of these breakdown influences is carried out either in that the parameters influencing the breakdown value are changed and/or in that

the breakdown values are copied by virtual breakdown values and are compensated with the help of these virtual breakdown values.

In both cases the breakdown value is corrected or compensated so that the breakdown influences or the breakdown values are eliminated or reduced to a permissible amount. By the virtual breakdown values copying the breakdown values the correct cause for a breakdown cannot be localised conclusively but the effect of the breakdown value on the overall system can however be positively influenced in the above sense.

Furthermore Figure 3 shows a block circuit diagram of a torque control with adaptation and the interaction thereof with any path and a setting member. The torque control described in the following can thereby be used for systems, such as torque transfer systems, with or without load distribution.

In the adaptation block 7 the adaptation of the secondary consumers is carried out. The secondary assemblies, such as eg dynamo, steering pump or climate control represent a branch of the torque and/or load flow by a part of the drive torque M_{mot} supplied by the engine being taken up by the relevant assembly. For a clutch control this means that it starts from a jive torque M_{mot} which is not really available, ie that the ideal clutch torque arising from the supposedly higher engine torque and thus also the setting value thus detected are too great. The detection of such a load distribution which is furthermore designated with adaptation of the secondary consumers can occur for example in that corresponding additional signals of measured values, such as the switching on or switching off of the climate compressor, climate control unit and

other secondary consumers are evaluated.

In a second adaptation block 9 a correction is carried out of interferences which can be caused by measurable values, such as eg temperatures - eg the cooling water temperature has an effect on the engine torque - or speed - the friction value can be changed over the slip. These corrections are furthermore designated with adaptation 1. A compensation and/or correction can in this case be carried out either through parameter adaptation, eg friction value correction in the further compensation block 28 or in the transfer block 30 through the temperature or however can be carried out through a system adaptation in the form of theoretically or empirically established breakdown models, eg a non-linear correction of the engine torque over a temperature.

In the third adaptation block 11 interferences which can be caused through non-measurable system input values and/or ageing and/or mean variations, are corrected and/or compensated. Since this type of interference, such as eg ageing or mean variation cannot be detected from directly measurable input values, it must be detected by observing system reactions. This means that such interferences cannot be compensated by remonstrances before they occur, but the reaction of the system as a deviation from the expected behaviour must be observed and then must be corrected and/or compensated.

These deviations can either be measured directly, eg by means of a torque sensor on the clutch or however they can be calculated from other measured values by means of a method model. In the event of detection corresponding reference characteristic fields or clear reference values of the system are required. In order to compensate an

interference detected in this way it is then necessary either to localise and correct the source of the interference or however for example a virtual breakdown source A or B is assumed at which the detected deviation is corrected. Similarly an interference can also be attributed to an existing block, such as eg the engine block 13 or the inverse transfer function of the transfer unit in the transfer block 30.

The ascribing of the interference can be frictional without such a block being responsible for the interference. The detection of the condition values, in contrast with the regulation, need therefore not be carried out permanently and can be reduced to certain operating areas.

In phases where there is no adaptation the adapted parameters are used which were detected in an earlier adaptation phase.

According to Figure 3 the drive torque 15 M_{mot} of the drive assembly 16, such as eg of an internal combustion engine, is formed and/or calculated in the engine characteristic block 13 from the different input values 14.

The values used to this end compromise at least two of the following values, namely the speed of the drive assembly, the load lever position or the accelerator pedal position of the fuel supply, the under pressure in the suction inlet system, the injection time, the consumption etc. Furthermore when forming or calculating the drive torque M_{mot} 15 it is possible to method the detections obtained in the possible breakdown practice (where, temperature).

In the connection block 17 a connection is produced which

as a result of taking into account the secondary consumers in the adaptation block 7 causes a correction of the drive torque. This correction is carried out on an additive path so that the forked moments of the secondary consumers detected in 7 are subtracted from the engine torque 15 M_{mot}. This corrected engine torque is designated in the following by M_{Netto} 18.

The engine torque 18 which is corrected by the forked torques of the secondary consumers is the input value for the block 19 which serves as the compensation block for the breakdown value correction or compensation. The breakdown sources whose breakdown values can be or can be compared with the breakdown values actually occurring can be simulated in the compensation block 19 by corresponding correction values or correction measures. The virtual breakdown values return to the adaptation block 9 and compensate the deviations and/or fluctuations in respect of the desired ideal state which (deviations and/or fluctuations) occur in the system as a result of for example manufacturing tolerances, contamination etc.

The correction can hereby be carried out through additive, multiplicative, functional and/or non-linear proportions. It is thereby generally only of importance that the effect of the interference is compensated or reduced to a permissible amount within an area of acceptable boundary limits. Thus for example additive interferences in the form of a virtual consumer can be considered and thus superimposed on the drive torque even if the interference has a different physical cause.

In the dynamic block 20 the dynamics of the method to be controlled, eg in the form of considering the mass inertia moments, eg of the moved engine mass, can be controlled

afterwards if this is advantageous for the behaviour of the system or for the control. Thus for example with severe accelerations or delays improvements are produced regarding the quality of control. The dynamic-corrected drive torque 21 is also designated M_{AN} in the following.

In the operating point detection block 22 the ideal clutch torque M_{KSoll} is fixed in dependence on the relevant operating point. This is calculated from a percentage proportion of the dynamically corrected torque M_{AN} and a safety torque M_{Sicher} which is described in the safety block 25. The percentage proportion is fixed through the torque division factor K_{ME} in a further characteristic field block 23. The percentage proportion of the dynamically corrected torque can be changed through a further correction block 24.

For systems with a genuine load distribution, such as in the case of a convertor with lock-up clutch, the proportion of the safety function can be $M_{Sicher} = 0$ since a torque is built up through the convertor in the event of slip.

In the case of an overall system without load distribution it must be insured through the safety function M_{Sicher} that for example in the event of slip an additive torque is added to the existing torque and thus prevents too much slip value from building up.

The correct proportion factor K_{ME} for each operating point is fixed or determined in the characteristic field block 23. This factor K_{ME} is recorded or stored in corresponding characteristic fields or characteristic lines in which one or more of the following values, such an engine speed, engine torque, driving speed etc is/are entered. This K_{ME}

factor represents in the case of two systems with a load distribution in the manner of a convertor with lock-up clutch the ratio to be set by the control between the transferable clutch torque and the shaft torque which is available.

In the case of systems without load distribution the direct proportion of the torque control is fixed by the proportion factor K_{ME}. The remaining torque is transferred through the slip-dependent safety torque which is detected in the safety block 25.

In the correction block 24 a further dynamic correction and/or compensation is carried out to the previously detected percentage proportion of the torque. This correction and/or compensation can be carried out in the manner of a restriction in the rise of the ideal torque and is designated gradient restriction in the following.

The gradient restriction can be carried out for example in the form of a maximum permissible increment per scanning step or through a predetermined time behaviour. Through this measure the excitation of the drive train is restricted to a maximum permissible amount and a good comfortable load change behaviour is thereby achieved.

In the safety block 25 a safety torque M_{Sicher} is determined at each operating point. This safety torque can be calculated for example in dependence on the slip speed. In this case the safety torque would be greater with rising slip. In the case of systems without load distribution the clutch can hereby be protected. Furthermore through a safety function of this kind it is possible to prevent or reduce thermal overloading of each relevant transverse system. The functional dependence between the safety

torque and the slip can thereby be described through a corresponding function or can be preset through characteristic lines or characteristic fields. The output value 27, the clutch ideal torque, of the superposed block 26 can be shown by

$$M_{KSoll} = K_{ME} * M_{AN} + M_{Sicher}$$

wherein the dynamic block 24 is not considered in this formula. Taking into account the block 24 the clutch ideal torque can be described as

$$M_{KSoll} = f_{Dyn} (K_{ME} * M_{AN}) + M_{Sicher}$$

wherein $f_{Dyn} (K_{ME} * M_{AN})$ involves the dynamic correction or dynamic consideration in block 24.

The clutch ideal torque is determined through the values of the torque division factor K_{ME} and safety torque M_{Sicher} 25 which are dependent on the operating point 22.

In a further compensation block 28 it is again possible through a second virtual breakdown source B to make a correction to the clutch ideal torque M_{KSoll}.

In the transfer block 30 this corrected clutch ideal torque M_{KSoll} core 29 is converted by an inverse transfer function of the transfer unit of the setting member into the setting value. By means of this setting value the transfer unit with the setting member 31 is controlled which then carries out the corresponding actions.

By transfer unit with setting member marked 31 is meant inter alia systems having a load distribution, such as converter with lock-up clutch or systems without load distribution in the form of a clutch, such as eg friction clutches. The clutches used in the cases of systems without load distribution can be for example wet-type clutches, dry-type clutches, magnetic powder clutches,

turning set clutches, safety clutches etc.

The production of the energy/force required to operate the setting member can thereby be carried out for example through electro-motorised, hydraulic, electro-hydraulic, mechanical, pneumatic or other ways.

Figure 4 shows a block circuit diagram of a control method with adaptation wherein the encroaching control block 5 and individual adaptation blocks are shown. The block 4 of the control path (not shown in this diagram) with setting member from Figure 3 also applies to Figure 4 and can be transferred from Figure 3.

Starting from the characteristic field block 13 an engine torque 15 is provided which is processed additively with a correction torque 42 so that the correction torque 42 is subtracted from the engine torque 15. The difference torque 43 is likewise additively corrected by the forked moments of the secondary consumer 7 wherein again the moments of each relevant secondary assembly are subtracted from the torque 43 corresponding to its state.

The moments are torques of the secondary consumers or secondary assemblies thus treated are determined or calculated from data or signals of the operating point 22 of the individual assemblies and/or from additional signals 44, such as eg switch-on and/or change-over and/or switch-off signals or typical operating signals, such as eg current voltage signals of the dynamo.

The detection can be carried out for example in that typical operating signals are recorded in a characteristic field or a characteristic line and thus an associated torque pick-up of the secondary consumer is determined by

reading a characteristic field or a characteristic line. A method of detection which is also possible is to store equations or equation systems where the signal values are entered as parameters and solving these equations or equation systems determines the torque pick-up.

The corrected signal 45 can undergo a dynamic correction through the dynamic block 20. The dynamic block 20 considers for example the inertia moments of the rotating components, such as engine parts and eg the flywheel or the inertia moments of other components of the drive train. The operating point 22 is determined or calculated from the condition values 40 of the system. This can be possible by determining data from characteristic fields or by solving the equations or equation systems wherein the condition values enter these equations as parameters.

From the operating point 22 the torque division factor K_{ME} 23 is detected for example from a characteristic field. The dynamic-corrected signal 46 is multiplied with the torque division factor 23 and thus the torque is determined which is transferred for example from a convertor lock-up clutch of a hydro dynamic torque convertor with convertor lock-up clutch. The signal can again be corrected by means of the dynamic block 24.

In the example illustrated in Figure 4, the dynamic block 24 is produced as the gradient restriction, ie a restriction of the maximum rise in the torque. This gradient restriction can thus be produced so that the rise in the torque as a function of the time within a fixed time area is compared with a maximum permissible value, such as eg a slope, an on exceeding the actual rise over the maximum value of the slope the slope signal is used as the real value.

A further possibility for a gradient restriction can be produced through a dynamic filter. The time behaviour of the filter can be selected differently depending on the operating point, so that when using for example a PT_1 filter the time constant can be set as a function of the operating point.

The output signal 47 of the block 24, the clutch ideal torque M_{KSoll}, is passed on, as according to Figure 3, to the transfer unit with setting member. This clutch ideal torque is compared with the clutch actual torque M_{KIST} 48 at the connection point 49. This comparison is guaranteed through an additive method in which the clutch actual torque is subtracted from the clutch ideal torque and thus a difference _M 50 is formed. The difference torque _M is processed in the following blocks of the block circuit diagram into a correction torque 42 which is processed at the connection point 52 with the engine torque 15.

The adaptation in this example of Figure 4 does not carry out any localisation of the breakdown values but traces the interferences to fictional breakdown values or interferences. The correction and/or compensation of these real breakdown values by means of fictional breakdown values no longer demands the localisation and correspondingly also no longer the correction of the real causes of error and errors themselves. With the example shown in Figure 4, the engine torque or engine characteristic field is regarded as the fictional breakdown source so that all faults occurring and interferences are signified as interferences of the engine torque and are compensated or corrected by an engine correction torque M_{mot_korr}.

The aim of the adaptation is to bring about the most accurate possible setting of the torque division factor K_{ME} in order to be able to produce an optimum reaction on the interferences and thus in order to be able to optimise the physical behaviour of the system.

The correction value M_{mot_korr} can be determined by solving the equations or equation systems and/or by using a correction characteristic field. The correction characteristic field can be produced so that the correction value is recorded eg over two dimensions. When determining the correction characteristic field it is possible for example to use the same dimensions through which the engine characteristic field is recorded, such as eg the consumption and engine speed. However it is also possible to use as a dimension of this correction characteristic field a value which reflects a dependence of the transfer function of the path, such as eg the turbine speed.

The design of such a correction characteristic field over the engine torque and the engine speed can be carried out for example through fixing three supporting points. Through three supporting points it is possible to fix a plane which determines the correction characteristic field as a function of the two dimensions. A further possibility exists where four supporting points are chosen and the four supporting points define a surface which determines the correction characteristic field. The block 51 carries out in this context an evaluation of the supporting points as a function of each relevant operating point. This evaluation of the supporting points is carried out since a statement on the correction values as in other operating points can be made over the surface of the correction characteristic field from each operating

point. Since this can however lead to errors and the statements in partial areas of the correction characteristic field cannot be linearly transferred to other partial areas, the evaluation of the supporting points is introduced.

The result of this evaluation is that depending on the relevant operating point or area of the operating point the supporting points are evaluated differently and thus the influence of points in the correction characteristic field which are further removed from the operating point has a lesser or greater significance. The evaluation of the supporting points is followed by a block 53 which affects the time behaviour of the adaptation. The block 54 represents the correction characteristic field block which determines from the operating point 22 the engine torque correction value 42 which is processed at the connection point 52 with the engine torque 15.

Figures 5a to 5c show in a diagrammatic illustration the possible interferences of the engine torque as a function of the time. In Figure 5a the ideal torque is shown as a horizontal line and the actual torque is shown as a horizontal line with a step. This step can be identified as an additive proportion of the engine torque which is caused for example through additional assemblies. A step in the actual torque is thus formed when switching on or switching off or switching over an additional assembly into another operating area. Depending on whether the forked load is increased or reduced, the step can increase or lower the actual torque. From the height of the step and from the time behaviour it is possible to obtain a statement as to which extra assembly was switched on, switched off or switched over.

Figure 5b shows the ideal torque and the actual torque in an operating state which is different compared to Figure 5a. The difference between the two curves can be identified as a breakdown with affects a multiplicative proportion of the clutch torque. A compensation and/or correction of this breakdown value must thus support a multiplicative character.

Figure 5c in turn shows the ideal and actual torque wherein the two moments are separated from each other by an additive proportion. The correction and/or compensation of this interference can be undertaken through an additive proportion of the clutch torque. The example in Figure 5b can be explained eg through a change in the friction value and the example of Figure 5c can be explained through a deviation of the setting value.

Figure 6 illustrates a correction characteristic field wherein the engine correction torque is shown as a function of the engine torque and the engine speed. More particularly, the four corner points of the value area are used as the supporting points 55. The evaluation of the supporting points in the block 51 of Figure 4 can be carried out for example in that at a certain operating point the supporting points are changed in their vertical position so that the close area around the operating point experiences a greater evaluation. This evaluation through a change of the vertical position of the supporting points can be designed depending on the operating point so that one to four supporting points experience this change.

The fixing of the four supporting points 55 which form a surface can also be modified in that six supporting points 55 are used, see Figure 6a, wherein there are always three supporting points arranged along one axis and through six

supporting points two surfaces are defined, each with four supporting points wherein two supporting points are claimed by two surfaces.

A further development can be characterised in that nine supporting points are used, see Figure 6b, in order to define four surfaces. The characteristic field is constructed so that each two adjoining supporting points belonging to one surface are connected by a straight line so that the perimeter of such a surface within the definition area is formed by four straight lines and the projection of the characteristic field to the definition area represents a polygon, more particularly not a rectangle nor a square. The connecting lines between two opposing boundary straight lines of the characteristic field which lie in one plane which is spanned by a straight side line of the characteristic field and the definition area axis of the characteristic field are likewise straight line sections.

A further development of the characteristic field of Figure 6 can produce a curved surface which is created according to a functional connection in the three-dimensional space, such as eg a parabola of the second order. The surface which characterises the characteristic field can be a curved surface which is defined by certain supporting points and/or a functional connection or an equation or an equation system.

Figure 7 shows a block circuit diagram or flow or development diagram of a torque control with adaptation of a torque transfer system which will be explained in further detail below. The torque transfer system can be for example a clutch, such as friction clutch and/or starting clutch of an automatic gearbox and/or a transfer

means of an infinitely adjustable cone pulley belt contact gearbox and/or a hydrodynamic torque converter with converter lock-up clutch and/or a turning set clutch and/or a safety clutch. The operation of the torque transferring parts can be carried out through an electro-mechanical, an electro-hydraulic and/or mechatronic and/or a mechanical and/or a hydraulic and/or a pneumatic setting member.

In accordance with Figure 7 the drive torque 62 of the drive assembly 61, such as in particular internal combustion engine, is first calculated from different input values 60. The values used here comprise at least two of the following values, such as speed of the drive assembly, load lever position or accelerator pedal position of the fuel supply, under pressure in the suction intake system, injection time, consumption etc. The drive assembly is shown in block 61 and the drive torque of the drive assembly at 62. The block 63 represents the connection which causes a correction of the drive torque. This correction is made by means of correction factors which are supplied from the system adaptation 64. This system adaptation 64 can be designed as a programme module which as a result of additional input values 65 analytically or numerically determined values and values of characteristic line fields undertakes a correction of the average drive torque. These correction factors can compensate the deviations which occur in the system from the desired state, namely by compensating these deviations through additive, multiplicative and/or non-linear proportions.

The block 66 represents the determination or fixing or calculation of a torque division factor K_{ME} which is correct for each relevant operating state and which lies

as a rule between 0 and 2. However system conditions can also occur which make it necessary to use a larger K_{ME} factor. This K_{ME} factor represents the torque ratio $M_{kupplung}$ to $M_{Antriebe\text{-}korrigiert}$ to be set by the control as one of the values previously fixed for each operating point in the manner of a characteristic field from the relevant selected evaluation of the criteria indicated in Figure 2, ie the K_{ME} factor is recorded in a characteristic field for the individual operating states.

The K_{ME} factor can however also apply as a constant in the entire operating area. Fixing or calculating the K_{ME} factor can also be undertaken through an equation or through an equation system wherein the solution to this equation or equation system determines the K_{ME} factor.

Condition values of the vehicle and the design of any possible torsion damper which may be present can be effected or considered in the characteristic field of the K_{ME} factor or in the analytical equations for determining the K_{ME} factor. The design of any possible damper, for example of a lock-up crutch, is thereby of particular importance since with the presence of such a damper the K_{ME} factor can be kept constant at least over a comparatively large section of the operating area of the internal combustion engine or of the hydrodynamic torque converter.

A K_{ME} factor which is kept constant over a wide operating area can also be effected for clutches, such as friction clutches or starting clutches.

The ratio of clutch torque and drive torque is fixed by the torque division factor K_{ME} factor. A moment-controlled slip operation is hereby possible for example. With systems have a load distribution (eg converter with lock-

up clutch) the torque proportioned which is to be transferred by the lock-up clutch is fixed by this factor. In systems without load distribution, eg clutch system, no less than 100% of the torque existing on the drive side can be transferred in a stationary operation. The factor in this case fixes which proportion is directly transferred through the torque control. The remaining torque proportion is controlled afterwards through a slip-dependent safety torque which copies a converter-like behaviour. At 67 the calculation of the ideal clutch torque is carried out by means of the relevant K_{ME} factor and the corrected drive torque of the drive assembly. At 68 a further correction of the ideal clutch torque can be carried out by the additive, multiplicative and/or non-linear proportions resulting from the system adaptation 64. The connection 68 can thus be provided. A corrected ideal clutch torque is hereby obtained. In many cases it is sufficient if only one of the two connections 63, 64 is present wherein preferably the connection 63 is to be maintained.

At 69 the calculation of the setting value is made from the corrected ideal clutch torque of the inverse transfer function of the path which represents the lock-up clutch or clutch. The block 70 represents the inverse transfer function of the setting member which comes into use in order to calculate the setting value which is required for the setting member 31. The setting value thus acts on the control path 72 which in turn acts on the vehicle 73. The value set by the setting member can be attributed to the control device in order to increase the control quality of the control method. This can hereby be for example the position of the transmitter cylinder of a hydraulic system set by the electromotor of an electrohydraulic setting member. This return takes place in the blocks 74 and 75.

The block 76 represents a calculator unit which serves to simulate a model of the vehicle and the torque transfer device.

Block 77 represents the measured value release of condition values of the vehicle which are processed as input values at another place in block 78.

The broken line in Figure 7 represents the transition area between the central computer or control unit and vehicle. At 70 the regulator output value can be calculated which is formed on the basis of the setting value detected at 69 and the inverse transfer function of the setting member. The setting member can be formed in a particularly advantageous way through an electrohydraulic or electro mechanic setting member. Advantageously a proportional valve can be used or a pulse-width-modulated valve.

At 75 a feedback of the setting value can take place in the form of a regulation or adaptation. This feedback can however also be dispensed with. At 79 a measurement of the actual clutch torque can be carried out, eg through a torque sensor or through an expansion measuring strip (DMS).

Instead of the measurement of the actual clutch torque which is carried out at 79 it is also possible to undertake a calculation of this torque from the condition values and from the vehicle and convertor physics. To this end for example the engine characteristic field and/or the convertor characteristic field or values representing the fields can be processed in a processor or in a central processor unit and/or also stored in a memory. Furthermore it is also possible to store for this purpose a characteristic field representing the torque

transfer capacity of the eg convertor lock-up clutch or a value representing same.

If a detection of the actual clutch torque is carried out according to points 79 and 76 it is possible to balance the measured actual clutch torque with the actual clutch torque calculated from the model. Balancing can thereby take place as a logical connection, eg on the minimum-maximum principle or as a plausibility comparison. The following comparisons inter alia can take place and the corresponding corrections can be made in the system adaptation characterised by 64 in Figure 7.

A: Comparison of the corrected ideal clutch torque and the actual clutch torque wherein this comparison can also be made long term, eg by observing the deviation through a co-rotating time window. The comparison between the corrected drive torque and the recharged drive torque can be formed wherein this comparison can also be carried out long term, eg by observing the deviations for a co-rotating time window. Likewise an evaluation of additional signals can be carried out, such as eg switching on or off of the additional assemblies, such as eg climate control, compressor etc, gear change.

B: Detection of the system deviation determined under A into additive, multiplicative and/or non-linear proportions of $M_{Antrieb}$ and $M_{Kupplung}$ and the resulting division into the corresponding adaptation loops 80 and 81 or into the connections 63 and 68.

The detection or determination of the corresponding proportions of $M_{Antrieb}$ and/or $M_{Kupplung}$ can be carried out for example according to the three diagrams of Figures 5a to 5c.

Figure 7 shows a development diagram of the control method with the individual processing steps. In a first processing step a drive torque of the engine is determined from a plurality of input values. There follows a first correction of this value according to the provisions of a system adaptation. This system adaptation is a program module which undertakes a correction of the average drive torque as a result of the additional input values, analytically determined values and characteristic line fields. In a further processing step this corrected drive torque is multiplied with a proportion factor K_{ME} which can lie between zero and two. This proportion factor K is recorded in a characteristic field for the individual operating conditions. In this characteristic field it is also possible to record condition values of the vehicle and the design of any torsion damper which may be present. The ratio of clutch torque and drive torque is fixed through this proportion factor K_{ME}. A controlled slip operation is hereby possible for example.

In the case of systems with a load distribution (convertor with lock-up clutch) the torque proportion which is to be transferred through the lock-up clutch is fixed by this factor. In the systems without load distribution (clutch system without parallel connected convertor) in the stationary operation nothing less than 100% of the torque existing on the drive side can be transferred. The factor fixes in this case which proportion is directly transferred through the torque control. The remaining torque proportion is controlled afterwards through a slip-dependent safety torque which copies a convertor-like behaviour.

The ideal clutch torque obtained is again corrected in a

next processing step according to the system adaptation. A corrected ideal clutch torque is hereby obtained. Finally by means of an inverse transfer function of the control path a setting value is determined from this corrected ideal clutch torque. By using the inverse transfer function for the setting member the value arising at the output of the control device is obtained from this setting value. This starting value is passed on to the setting member which in turn acts on the control path and the vehicle. The value set by the setting member can be returned to the control device to increase the control quality of the control method. This can be for example the position of the transmitter cylinder set by the electric motor. Furthermore additional system values, such as for example the clutch path, or vehicle values can be sent to the control device. These additional input values are then entered into the described control method through the system adaptation.

Figure 8 shows a simple model of an adaptation which is restricted to the additive correction of the drive torque. The deviations which result from the difference between the ideal and actual clutch torque are adapted through virtual breakdown sources. Figure 8 thereby shows with block 71 the drive assembly, such as internal combustion engine, which produces an engine torque 62. The block 90 represents the adaptation by means of virtual breakdown sources, whose output signal is processed at the connection block 91 in an additive way with the engine torque 62. The corrected engine torque is corrected dynamically in block 2 by means of dynamic corrections based on the inertia moments of the flywheel.

The torque arising for example on the torque convertor with lock-up clutch is divided into two proportions by

means of the torque division factor whereby one proportion is transferred by the lock-up clutch 3b and the difference torque between the torque transferred by the ensuing torque and the torque transferred by the lock-up clutch is transferred by the torque convertor 3a.

Figure 9 shows a block or flow diagram of a control method for torque transfer systems, the broken line in the lower half of the figure representing the separation between the central computer unit and the vehicle. The control method of the block circuit diagram shown in Figure 9 represents a simplified design of adaptation. The control of the lock-up clutch is thereby carried out electro-hydraulically through a proportional valve or a pulse-width-modulated valve. The output signal of the regulating computer or the computer output value is a setting current which is set in proportion to a scanning ratio adjoining the for example pulse-width-modulated output of the computer. The clutch torque results for example from the pressure difference controlled in this way at the convertor lock-up clutch or between the two pressure chambers of the lock-up clutch. The system adaptation is restricted to the adaptive correction of the drive torque whose deviation results from the difference between the ideal and actual torque.

With an embodiment of the control method according to Figure 9 the connection 68 or the return of the corrected drive torque (M_{ANkorr}) is omitted compared with Figure 7. In Figure 9 the ideal pressure difference DP_{soll} is determined at 100, namely as a function of the ideal clutch torque as the main value and where applicable still in dependence on the corrected drive torque M_{ANkorr} and the turbine speed N-Turbine as parameters.

The additional function block 101 according to block 70 of Figure 7 is divided in Figure 9 into two sub-function blocks, namely into 101a and 101b. A feedback coupling 102a and 102b is allocated to each sub-function block 101a and 101b. The input value of the inverse transfer function of the setting member (101 = 101a and 101b) is the ideal pressure difference (DP_{Soll}) calculated in block 100. The output value is formed through the associated scanning ratio as the regulator output value.

The adjoining setting member is divided into the electrical setting member proportion which is formed by an end phase and the valve winding, as well as into the hydraulic setting member proportion which is decisive for the corresponding pressure biasing of the convertor lock-up clutch, see block 103. The input value of the electrical setting member proportion is the scanning ratio. This is converted on the output side into an actual current. In dependence on this actual current (I-Ist) the hydraulic setting member proportion sets a corresponding pressure biasing of the convertor lock-up clutch. This is carried out by setting a corresponding pressure difference between the chambers of the convertor lock-up clutch.

Block 101a represents the inverse function of the hydraulic setting member proportion in that the associated ideal current is calculated from the ideal pressure. This proportion of the setting member has a feedback of the measured actual pressure in the form of a pressure adaptation which is illustrated through the block 102a. This pressure adaptation 102a supplies the corrected ideal current. The second part 101b of the inverse transfer function 101 of the setting member represents the electrical proportion which calculates the associated

scanning ratio from the corrected ideal current. A PID regulating algorithm is used for this. The input value I_{Soll-R} for the inverse transfer behaviour of the electrical setting member proportion is thereby calculated from the control deviation $I_{Soll-Korr} = -I_{Ist}$ (I_{Ist} is measured according to the valve winding) with a PID regulator.

The numbering of the individual blocks selected in Figure 9 substantially corresponds to the numbering of the individual blocks of Figure 7. In this way the individual function blocks of the special electro-hydraulic design according to Figure 9 can be related to those of the general design according to Figure 7.

The individual markings contained in Figure 9 have the following significance:

DP_{Soll} = 110 = Ideal pressure difference at the lockup or convertor lock-up clutch. Corresponds to the pressure difference between the pressures prevailing in the chambers existing either side of the piston.

DP_{Ist} = 111 = Actual difference between the two chambers of the convertor lock-up clutch.

P_{Nach} = Pressure after the lockup or convertor lock-up clutch.

I_{Soll} = 113 = Ideal current for the electro-hydraulic valve.

N = 114 = Speed difference between pump wheel and turbine wheel, thus N = N pump wheel - N turbine wheel.

The condition values of the vehicle 115 indicated in Figure 9 in front of the block marked 76 retain the slip

in the lock-up clutch or in the convertor.

As can be seen furthermore from Figure 9 the speed difference N = N pump wheel - N turbine wheel does not represent any regulating value, as is the case with the known slip regulations. With the torque control according to the invention this speed difference N is used as a condition value of the path to be controlled to observe possible torque deviations which then have a correcting effect on the control in the adaptation through corresponding connections. The observed torque values can hereby be stored, eg in the manner of a co-rotating time window over a certain time period in order to detect the proportions of deviations at the clutch and engine. This takes place in the system adaptation marked 116.

The control according to the invention furthermore has the advantage that the adaptation of the breakdown proportions of the drive torque can also take place with a completely opened lock up or convertor lock-up clutch, thus with $K_{ME} = 0$. To this end the nominal drive torque is compared with the torque adjoining the convertor, which is carried out in the connection 63 of Figure 7 or with the processing step 63 of Figures 7 and 9. Through this adaptation in anticipation of a later closing of the lock-up clutch possible deviations of the drive torque are already considered in the open state of the lock-up clutch. For this in the system adaptation 116 or 64 the torque adjoining the convertor is determined, namely preferably the convertor characteristic field is recorded or stored in this system adaptation. By determining the speed difference between the turbine and pump wheel it is thereby possible to determine the ensuing torque. This convertor torque is then compared with the nominal drive torque of the engine or drive assembly. This drive torque

can be derived from a stationary engine characteristic field recorded in the block 61 according to Figures 7 and 9, namely as a result of the measured condition values, such as more particularly engine speed, load lever position, consumption, injection amount, injection time and so on. The speed difference between the turbine wheel and pump wheel can be determined in the block 76.

Furthermore it is possible to determine the convertor torque already in block 76 wherein then the convertor characteristic field is recorded in the block 76.

Figure 10 shows a vehicle 201 with an internal combustion engine 202, which acts on a gearbox 204 through a clutch 203 which is self-adjusting or adjusts to the wear. The gearbox 208 is connected through a drive shaft 205 with a drive axis 206 of the vehicle 201. With the self-adjusting clutch 203 or clutch which adjusts to the wear a difference is made between a drive side 207 adjoining the combustion engine 202 and an output side 208 facing the gearbox 204. To the engagement or disengagement system of the clutch 202 is attached a transmitter cylinder 200b which is connected through a hydraulic pipe 209 with a transmitter cylinder 211. The engagement or disengagement system, such as mechanical disengagement bearing can enter into contact with the plate spring tongues of the plate springs so that the force biasing of the clutch plate spring is determined opposite the pressure plate which biases the pressure plate in the direction on the engine side and thus biases the friction linings between the pressure plate and the flywheel. The hydraulic pipe 209 is connected through a transmitter cylinder 211 with an electric motor 212 wherein the electric motor 212 and the transmitter cylinder 211 are combined in a housing to one setting member 213. A clutch

path sensor 214 is mounted in the same housing directly adjoining the transmitter cylinder 211. Furthermore a control apparatus not shown in the drawing is mounted on a conductor plate 227 inside the setting member housing. This electronic control device contains the power and also the control electronics and is thus mounted completely in the housing of the setting member 213.

The control apparatus is connected to a throttle flap sensor 215 mounted directly on the combustion engine 202, a motor speed sensor 216 and a tacho sensor 217 mounted on the drive axis 206. Furthermore the vehicle 201 has a gear lever 218 which acts through a switching rod on the clutch 203. A switching path sensor 219 is provided on the gear lever 218 and is likewise in signal connection with the control apparatus.

The control apparatus provides the electromotor 212 with a setting value in dependence on the attached sensor system (214, 215, 216, 217, 219). For this a control program is implemented in the control apparatus either as hardware or software.

The electric motor 212 acts on the self-adjusting clutch 203 in dependence on the allowance of the control apparatus through the hydraulics (209, 210, 211). The function of this clutch 203 has already been described in detail in the published specifications DE-OS4239291, DE-OS4306505, DE-OS4239289 and DE-OS4322677. The contents of these specifications is referred to explicitly as belonging to the circumference of the disclosure of the invention. The advantage of a self-adjusting clutch 3 is that the forces necessary for operating the clutch are clearly reduced compared to conventional clutches as a result of the wear-adjusting construction. Thus the

electro motor 212 can be dimensioned with lower power take-up and power discharge and thus the setting member 213 can be made more compact overall. The setting member 213 in Figure 10 is not shown to scale compared with the other components of the vehicle 201.

The setting member 213 is explained in further detail with reference to Figures 11a, 11b and 12a, 12b. The electric motor 212, more particularly the direct current motor, acts through an engine shaft 220 on a worm which meshes with a segment wheel 222. A crank is fixed to the segment wheel 222 and is in active connection through a piston rod 224 with the piston cylinder 225 of the transmitter cylinder 211. A snifting member 250 with snifting bore 251 is formed on the transmitter cylinder 211 to compensate thermal influences on the hydraulic fluid.

The electric motor 212, such as a direct current motor biases through the gearbox which can be self-locking, the hydraulic transmitter cylinder 211 with pulling or pushing forces. These forces are transferred through the hydraulic pipe 209 to the clutch 203. The clutch 203 is hereby engaged or disengaged in controlled manner.

Since the parallel axes of the transmitter cylinder 211 and the engine shaft 220 are arranged in different planes, thus are offset, the space taken up by the setting member 213 is considerably less.

A servo spring 226 is provided concentric with the axis of the transmitter cylinder 211 inside the cylinder piston 225 or inside the transmitter cylinder housing 211. This servo spring 226 supports the electro motor 212 during the disengagement method of the clutch. During the engagement method of the clutch, the spring is tensioned by

overcoming its force action.

The interplay between the electric motor 212 and the spring 226 is explained with reference to the diagrams illustrated in Figure 13. The force paths are each entered over the clutch path. The solid line 237 represents the force applied by the electro motor 212 during the engagement and disengagement method of the clutch wherein the upper line signifies the force path during the disengagement method and the lower line during the engagement method. This force path shows that the disengagement method requires higher forces than the engagement method. The chain dotted line 239 is the spring characteristic line of the servo spring 226. The broken line 238 shows the interaction of the forces of the spring 226 and the electro motor 212.

The force 238 to be applied overall by the electro motor 212 is clearly reduced, as shown by the displacement of the broken force line in the direction of smaller forces. Through the supporting action of the correspondingly selected servo spring 226 the characteristic line of the electro motor or plate spring is moved in the negative force direction and the maximum amounts detectable in Figure 13 in the positive and in the negative directions of the broken line are approximately equal. Through this supporting action of the servo spring 226, it is possible to make the electro motor 212 correspondingly small than in comparison with the sizes without the support of the servo spring 226. The support of the servo spring in this way likewise assumes that the electro motor is used in the pull and push directions.

In Figure 12a a servo spring 226 is mounted in the actor housing wherein it is housed between two contact bearing

areas 227a, 227b. The contact bearing area 227a is biased through spring tension against a spring ring 228 which is connected to the piston rod whilst the contact bearing area 227b is supported on an area of the actor housing. In order to protect the gearbox against dirt a rubber membrane 229 is arranged in the area of the contact bearing area 227a. Furthermore, the housing has a ventilation bore 230 which allows drainage in the event of hydraulic fluid emerging.

The operating method of the control method implemented in the control apparatus for the torque control of a torque transfer system, such as a friction clutch, is shown in simplified form in Figure 14. The control method is stored as a software program in a for example eight bit processor of the control device. The electric motor 212 for example can be controlled with this control method. By means of the throttle valve sensor 215 and the engine speed sensor 216, a drive torque M_{Mot} of the engine 202 is determined and made available to the control program as the input value. The engine speed sensor 216 detects an engine speed N1 and the tacho sensor 217 registers a speed of the drive axis 206, the additional input values are sent to the control program. A gearbox input speed N2 is calculated by means of the speed of the drive axis 206. The difference between the speeds N1, N2 is designated the slip speed. The slip speed is analytically determined within the control program and monitored for exceeding a slip boundary value. Exceeding the slip boundary value is detected as a slip phase S. This slip phase S stops until the slip boundary value is again understepped.

The clutch torque M_K is calculated by means of a correction value M_{Korr} according to the formula

$$M_K = M_{mot} - M_{korr}$$

The correction value M_{korr} is a torque value which is increased incrementally with the computer cycle and is reduced in the times detected as slip phases S according to the action of the control program. Through this method the clutch 203 is operated constantly about a slip limit R. The slip limit R is the time at which the engine speed N1 begins to exceed the gear input speed N2. This is exactly the case when the torque arising on the drive side is greater than the clutch torque momentarily transferable by the clutch. This method also functions then when the drive torque is not constant.

The characteristic line field illustrated in Figure 15 is evaluated before passing the setting value on to the setting member, more particularly in the case of a torque transfer system, such as a friction clutch.

The area of the possible setting member allowance, thus the area of the possible transferable clutch torque is entered on the abscissa. This area is divided into partial areas 240 of which one is shown shaded. This designated area 240 is the transferable clutch torque between 100 and 140 Nm. So long as the transferable clutch torque calculated according to the control method lies within this partial area, a permissible value of 140Nm is given to the setting member. In other partial areas 240, the procedure is similar.

Through this method, the number of setting movements of the setting member is further reduced. The setting movement, thus from one plateau to another plateau is fixed to a certain size. This design of the characteristic field regarding the setting movement can be such that the number of blocks or areas 240 can be different in dependence on the type of use. These

measures increase overall the life expectancy and lower the energy requirement of the actuator system of the torque transfer system.

Figures 15a to 15e show a setting member target carried out according to the control method for an ideal clutch torque.

By automating the clutch operation, an actor is required which allows the conversion of control signals into opening or closing processes or movements of the clutch. An adaptive control of the setting behaviour of the actor can be carried out so that a torque matching is effected. The use of a torque matching can advantageously lead to the setter undertaking not only the opening and closing processes during the gear changing and starting but set the clutch contact pressure during the entire driving operation so that the transferable clutch torque at each time corresponds to an ideal clutch torque resulting from the driving condition or operating point or a correspondingly desired excess contact pressure or lower contact pressure can be carried out in comparison to the clutch torque. This has the result that the setter during the gear changing processes need not move from the fully engaged position over the entire setting area in order to disengage the clutch, since as a result of the torque matching a setter position is already set which corresponds to the ideal torque actually set plus a desired offset value. Thus the demands on the dynamic behaviour of the system, more particularly of the actor, can be reduced regarding the design for maximum adjustment speed since as a rule shorter adjustment paths need be overcome or covered.

A dynamic torque matching designed in this way leads to

the actor with the electro motor having to be in operation during the entire operating time or driving time in order to be able to carry out a quasi instantaneous adjustment according to the dynamic changes in the actual torque.

With a control method which guarantees the torque matching at each time, an electro motor must constantly copy for example variations of the transferable torque. One possibility for using the electro motor only when required can lead to a copying of the clutch torque which is carried out in stages or in steps.

The control method must ensure at each time that an ideal clutch torque determined at each time step can be transferred through the clutch. The copying of the clutch torque has the effect that slight over pressures _m lying at a certain scatter band are tolerated, this means that matching movements and thus the load on the setting member can be reduced. The curve 241 of Figure 15a represents the calculated ideal clutch torque wherein the function 242 corresponds to the ideal clutch torque plus a scatter band. The values for the scatter band 242 are produced from the step height _M and the conditions that the set clutch torque may not exceed the calculated clutch torque and that a change in the set clutch torque is only carried out if the change exceeds a boundary value.

Figure 15b shows for example a method of operation according to a control method wherein the ideal clutch torque is adjusted above a boundary value 243 and for ideal clutch torque values less than equal to the boundary value the set clutch torque assumes a value which can be the same or different compared with the boundary value. By fixing the scatter band and a corresponding control, a definite over contact pressure takes place in some

operating areas which however leads to the action of the setter being reduced in time and the load of the setter thus likewise being reduced. The method according to Figure 15b shows that with low ideal clutch moments, the minimum clutch torque is set and thus the setter movements which are connected with a load on the setting system can be reduced. The minimum clutch torque 243 can for example be dependent on the operating point, such as for example on the transmission of the gear position, the engine speed, the accelerator pedal position or on a brake signal. Figure 15c shows a dependence of the minimum clutch torque as a function of the operating point, wherein the curve 244 is adapted in stepped manner to the dynamic behaviour of the operating point and the copied clutch torque 241 is adapted accordingly.

The method procedure illustrated in Figure 15d leads to a minimum clutch torque dependent on the operating point plus one according to the method of the stepped matching in relation to a scatter band of combined behaviour.

Figure 15e has a behaviour of the clutch torque which is predetermined by a minimum clutch torque 243 which cannot however be represented in areas with constant value, but is a function of the time wherein this minimum clutch torque is adapted through a step function 245 and for ideal clutch moments 241 which are greater than the minimum clutch torque a quasi instantaneous copying of the torque is carried out without undertaking an adaptation in relation to a scatterband.

Figure 16 shows the circuit diagram of a conventional H-shift. A difference is made between individual shift lanes 250 and a selection path 251 for the selection of the individual shift lanes 250. The path covered by the

gear lever 218 within the shift lanes 250 is designated the shift path 252. The directions of torque of the shift path 252 and the selection path 251 are indicated by corresponding arrows in Figure 16.

The position of the shift lever 218 can be detected by means of two potentiometers, such as in particular linear potentiometers. One potentiometer hereby monitors the shift path and a further potentiometer monitors the selection path. In order to carry out the monitoring method which can likewise be implemented in the control apparatus, the shift path and/or the selection path is detected and evaluated. The method of operation of the monitoring method is explained with reference to Figure 17. The signal paths which are relevant for the monitoring method are shown in Figure 17 in a diagram over the time t. The coordinate inscription corresponds to a division of any kind inside the computer of the shift path 252 detected. In detail, a gear lever signal 260 is registered over the time t which is proportional directly with the detected shift path 252.

The registered path of the gear lever signal 260 corresponds to a typical gear change method. The gear lever 218 remains in its position approximately until the time t marked here by 8.3 seconds. The gear lever signal 260 has up to this time only the vibrations which are typical in the driving operation.

These vibrations arise in the torque transfer system itself and are excited from outside additionally for example through unevenness in a roadway. After the time designated 8.3 seconds the gear level 218 is moved into the shift lane 250 so that the gear lever signal 260 rises from an approximate value of 200 increments to about 480

increments. This value remains constant for some time. This corresponds either to a stopping by the user or however to the time which is required for passing through a selection path 251. Finally a gear is engaged. The gear lever signal 260 rises to about 580 increments and remains approximately constant for sometime. This corresponds to the time interval for the synchronisation of the gearbox transmission to be engaged. The gear lever signal 260 then rises to a value which corresponds to the newly engaged gear.

In addition the gear lever signal 260 is filtered digital/analogue with an adjustable delay time so that a linearized filter signal 261 following the gear lever signal 260 is produced. The filter signal 261 is biased with a constant value and an offset signal which is dependent on the drive torque of the drive unit 202. The sum signal thus formed is entered on the diagram of figure 17 as the comparison signal 262.

The switching intent detection is carried out in dependence on the monitoring of the time dependencies of the parts on the gear lever signal 260 and comparison signal 262. As soon as the path of the gear lever signal 260 crosses the signal path of the comparison signal 262 a switching intention counter is set to zero and started. This time point is designated T_1 in the diagram. Furthermore the counting value of the switching intention counter runs in dependence on a computer cycle to a defined counting value highpoint. Here an accurately measured control time is provided in which the detected switching intent is verified. The counter can be stopped in this time by the control signals which occur and can be set again to zero. Such control signals can be conveyed by an attached sensor system. These sensors monitor

further influence values, such as the drive torque, attached load or the further movement path of the gear lever 218. As soon as this sensor system picks up measured values which contradict the detected switching intention a control signal is passed to the switching intention counter. The torque transfer system is hereby protected against faulty releases through the described monitoring method. Only when the switching intention counter reaches the defined counting value without a control signal having been sent is a switching intention signal sent to a secondary operating system.

The formation of the comparison signal 262 is explained in further detail with reference to figure 18.

The gear lever signal 260 is again shown on a different scale and the filter signal 261 produced herefrom. In order to form the comparison signal 262 the filter signal 261 is increased by a constant value and by an offset signal which is dependent on the drive torque. The constant value must be selected so great that the path of the gear lever signal 260 does not as a result of the typical operating vibrations of the gear lever 218 during operation of the vehicle intersect the path of the comparison signal 262 without a switching intention being provided thus leading to faulty releases. This must itself then apply if the drive torque, for example, by withdrawing fuel, has become zero and thus the offset signal has become zero. The time point for the withdrawal of the drive torque is here designated as T_2. Consequently the comparison signal 262 corresponds to an intermediate comparison signal 263 which is only composed by adding the filter signal 261 and a constant value. The constant value is advantageously adapted in operation to the elasticity of the gear shift rod and thus the potential

vibration width, such as vibration amplitude, of the gear lever.

Figure 19 shows the path of a gear lever signal 260 during an extremely slowly implemented gear change method. When the shift handling is carried out delayed there is the danger that the gear lever signal does not intersect the comparison signal. This would have the result that the existing shift intention is not reliably recognised. For this reason the monitoring method is additionally expanded by the monitoring shown here of the gear lever change, i.e. a change in the gear lever path as a function of the time. Thus the change in the gear lever signal 260 is monitored in that the path change established in a time window in a defined area outside the area which the non-operated gear lever occupies, is checked to see whether a boundary value is understepped. The understepping of this boundary value is recognised as a switching intention independently of the path of the comparison signal 262. In the case illustrated here the shift handling begins at a time point t_3. The monitoring area of the gear lever path extends from a first path s_1 to a second path s_2. The monitoring time window extends from a time point t_4 to a time point t_5. The path change established in this time _t within an area s understeps a stored boundary value and a switching intention signal is thus sent to the secondary operating systems.

The method of operation of the switching intention counter is explained with reference to Figure 20. In the example indicated here it comes at the time point t_5 to a peak in the path of the gear lever signal 260. This peak causes an intersection of the gear lever signal 260 and comparison signal 262. The switching intention counter is then started at time point t_5. At the same time as the

switching intention counter however a timer is started. When the peak in the gear lever signal path 260 swings back resulting in a renewed intersection of the gear lever signal 260 with the comparison signal 262 the said timer receives a signal. The timer is stopped and the time indicated is compared with a stored minimum time. In the present case it is established that the time detected by the timer lies below the stored time. In this sequence a control signal is sent to the switching intention counter. The switching intention counter is hereby stopped and reset to zero. Thus a switching intention has been recognised through the peak in the time point t_5 and consequently the switching intention counter has been started but a conveyance of the switching intention signal to the secondary operating systems has not taken place since a control signal was detected in the control time restricted by the starting of the switching intention counter. In contrast to this the switching intention actually existing at the time point t_6 is recognised and evaluated in the described way. Shortly after the time point t_6 a switching intention signal is sent to the secondary operating systems.

Figure 21 shows a diagrammatic illustration of a clutch operating system 300 for a vehicle. The total path considered thereby consists essentially of the system parts engine, setting member 301, such as e.g. an electro setter, connecting system 302 and a torque transfer system 303, such as a clutch.

The setting member 301 is designed as a mechanical or hydraulic or pneumatic setting member. The connecting system which is mounted between the setting member 301 and the torque transfer system 303 such as a clutch, can be a rod linkage in the widest sense or a hydraulic connecting

means. One embodiment of a hydraulic system is illustrated in figure 21 wherein a transmitter cylinder 304 is connected by a hydraulic pipe 305 to a receiver cylinder 306.

A device for power assistance can be mounted in the transmitter cylinder 304 and/or in the receiver cylinder 306. The power assistance device 307 can be designed for example as a coil spring or plate spring.

The torque transfer system 303, such as a clutch, can be a friction clutch and/or a self-setting clutch or a clutch, such as a SAC-clutch which compensates or automatically adjusts to the wear.

A control method with path adaptation of the clutch operating system is based on the fact that as a condition of a successful adaptation the individual system parts are examined for possible changes.

So that such an adaptation can be successful, it must first be explained which problems or which effects can affect the individual system parts and thus can influence an adaptation. For this reason the components mentioned above will be briefly dealt with again and principle sources of error and problem areas will be indicated.

The engine torque is generally determined or calculated by a characteristic field on the basis of the engine speed and the suction intake pressure (or as an alternative throttle valve angle). Similarly the solution of a same system can be used to determine the engine torque. Errors in the characteristic field and/or when determining the suction intake pressure can result in deviations in the actual torque. Furthermore the secondary assemblies in

its torque take-up are not known. This produces a further inaccuracy when determining the actual engine torque. Furthermore special features in the engine control (idling regulator, knocking control, pulling power switch off) can likewise lead to faulty results when determining the engine torque. An adaptation of these special features in the relevant engine control can be taken into account with an adaptation strategy in order to ensure a determination of the engine torque. In the case of the electronic systems provided for example for the pulling power switch off, signals can be processed for example which in relation to the pulling power switch off pass on a signal to the electronic clutch management in order to ensure the most accurate possible determination of the engine torque.

The setting member 301 can be designed as an electro-setter. A target of the ideal path, for example the clutch pressure plate, is converted in this system through a path-control or regulation. For a regulation the knowledge of the actual path is absolutely necessary in order to be able to regulate the system without permanent control deviation. The actual path can be measured and is thus available for further calculations. From the actual path and with a theoretical clutch characteristic line it is possible to calculate a theoretic actual torque M_{kistth} (it is thus not compulsory to use the ideal path and to come close to the time behaviour of the controls through a model.

A further way of obtaining an additional secondary value for the adaptation is to calculate a theoretical knocking force through the tension and resistance. By means of this knocking force it is possible to calculate a second theoretical actual M_{KIst2}. When changing the knocking force the changes in the clutch torque must be reflected. If

this is not the case then corresponding corrections can be carried out. A further possibility lies in using general forces for transfer wherein each relevant actual value of the forces can be compared with the corresponding value of the actual torque in order to determine whether an agreement of the counting value is established in the case of an engaged and/or disengaged clutch.

If a hydraulic system is used as a connection between the setting member and the clutch then the temperature of the system and the viscosity of the transfer medium play a decisive part. Similarly the lengths of the pipes and pipe cross sections can be taken into account since in the event of temperature changes and temperature differences these sizes can be subject to variation and may lead to inaccuracies. Similarly the connecting pipe between the receiver cylinder and the transmitter cylinder can be subjected to an expansion, such as a change in length or change in cross section whereby a false coupling position would be signalled.

The torque transfer system can be a clutch or a self-setting clutch. The so called influences are to be established in a change in the contact pressure forces or a change in the friction value. The changes which arise in relation to the contact pressure forces are described further below.

An adaptation can also designate the change of the friction value over the energy input and the change of the friction radius as function energy input.

An adaptation strategy can provide that the clutch torque is only adapted from a certain minimum value, see figure 22.

An adaptation of the complete setting system of the clutch actuator unit (comprising the engine, setting member, hydraulic system and a clutch) provides for an identification of the amounts of the individual system parts. Each system part is thereby analysed and the possible sources of error can be detected and the consequences of these possible sources of error can be estimated and removed or reduced. It can also be checked which sources or effects are important and which can be negligible.

The adaptation can provide additive proportions which are taken into account. By additive proportions are meant those proportions which are independent of the absolute value or the absolute level of the torque. The additive proportion can be taken up e.g. through secondary assemblies (consumers in front of the clutch). However faults in the engine torque characteristic field can also be compensated through additive proportions.

Figure 23 illustrates a diagrammatic model or a block circuit diagram which takes into account the additive proportion. In block 400 the engine is shown with its adjoining engine torque M_{an}. The block 401 shows the consideration of the additive proportions by e.g. secondary assemblies and faults in the engine characteristic field. The correction torque M_{Korr} to be introduced is considered at the connection 402, wherein:

$$M_{anKorr} = M_{an} - M_{Korr}$$

The inertia torque of the system is considered at block 403. This can mean that for example only the inertia torque of the flywheel or however of parts of the drive

train are considered. A dynamic-corrected torque is formed at 403 in order to determine the torque adjoining the clutch 404.

This torque can be corrected or adapted by a multiplicative proportion. Sources for the requirement of a multiplicative proportion are for example the changing friction value, eg as a function of the temperature and setting lining springs with their changed spring characteristic.

If the assumed and the actual friction value differ from each other then the error becomes greater the higher the required clutch torque.

The block 406 shows the vehicle mass in the block circuit diagram of Figure 23.

An adaptation method can be designed in that with a consumer adaptation it is ensured that the clutch torque ($M_{KSoll-Korr}$) is reduced so far that it leads to slipping of the clutch. This can be explained in that the value of M_{Korr} (correction of the assembly) is increased according to the equation

$$M_{KSoll-Korr} = K_{me} * (M_{an} - M_{Korr}) + M_{sicher}$$

until a slip is set. During this slip phase the clutch torque can then be increased again according to a pre-determined always accurately defined function (e.g. slope-shaped lowering of M_{Korr}) until the slip is reduced. From this behaviour an evaluation of the consumer can take place wherein the evaluation can be carried out each time or always once or several times per slip cycle.

In the ideal case that the actual clutch characteristic

agrees with the assumed characteristic line the value M_{Korr} contains the torque proportion which the consumers branch off or require. As a result of this estimation or calculation, taking into account an error in the engine torque, it is possible to provide details on the friction value.

Since no negative consumers appear negatively adapted consumers can be adapted or interpreted as a friction value which is too low. Furthermore the torque take-up of the individual consumers is restricted wherein each relevant absolute level need not be known. Exceeding a boundary value can thus be interpreted as a friction value which is too high.

Fixing an upper barrier or a boundary value can with skilled selection avoid the value being selected too great and the friction value alteration only being detected too late. It can likewise be avoided that when the boundary value is too low the secondary consumers are interpreted as a change in the friction value.

It can be advantageous if the adaptation is only carried out in the pulling-type operation wherein it should be carried out above a minimum torque.

This simple adaptation method, see also figure 14, has the result that a splitting-up of the adaptation model into an additive proportion (consumer etc) and a multiplicative proportion is only carried out by fixing or determining the limits. Within the limits the proportion is assumed as additive, and outside the limits is assumed as multiplicative faults of other causes, such as e.g. in the engine torque.

An error or a breakdown in the engine torque are added to the consumers or a clutch characteristic line.

Figure 24 provides an example for an embodiment, an estimation or appraisal of the additive and multiplicative proportions in the slip phases with different load conditions.

Line 450 shows the time path of the corrected clutch torque. Line 451 gives the time path of the engine speed n_{mot}, line 452 the time path of the gear box input speed n_{Getr}.

At the beginning of the observation time point shown in this example the engine speed 41 is approximately equal to the gearbox speed 452. The corrected clutch torque shows a slightly dropping time behaviour.

In the time period 453 a slipping phase takes place and the engine speed 451 lies slightly above the value of the gearbox speed. After the detection of the slip phase the clutch torque 450 is increased. In the time period 456 the engine speed 451 reaches a relative maximum and the increase in the clutch torque lets the engine speed drop again.

At the beginning of the time period 454 a tip-in takes place, i.e. a speed increase in the engine speed is briefly introduced. In this phase no adaptation takes place and the gearbox speed 452 follows the engine speed 451 with time delay.

The time period 455 shows corresponding to the time period 453 a slip phase.

Since the consumer adaptation drives or can drive permanently at the slip limit, there is the further possibility of evaluating the slip phases at which the overall contact pressure changes or has changed, i.e. the ideal moments on the clutch or on the torque transfer system lie at a different level, such as for example show a different engine torque and/or load conditions. A requirement for this is that the actual consumer has not changed, i.e. too long a time span between the slip phases appears as not very favourable.

If with different load conditions, such as in the slip phases 453 and 455 the consumer value does not change then it can be taken that the assumed and/or determined and/or calculated friction value corresponds to the actual friction value of the clutch.

In such a case the friction value can be corrected or a correction can be undertaken.

With this embodiment it is advantageous that a division into an additive and multiplicative proportion can be carried out.

In the event of a change in the consumer during the time of the adaptation a separation of a friction value change and a consumer change cannot be correctly carried out which can be substantially compensated through an increased frequency of the adaptation method.

Furthermore an adaptation can be carried out in the constant phase after load changes which as a result of possible long time intervals can be combined with other adaptation strategies.

An adaptation of the multiplicative proportion in dynamic areas or cases such as e.g. a tip-in and/or when starting off can likewise be carried out. In the event of slip the following applies

$$Man - Mkorr - \frac{\mu ist}{\mu theo} * Mksollkorr = J * d\frac{\omega}{dt}$$

By means of this equation it is possible to detect the unknown values wherein μ_{ist} and μ_{theo} are the actual and theoretical friction value.

This adaptation method is explained in further detail with reference to figure 25. Figure 25 shows the time behaviour of the ensuing torque 500, clutch actual torque 502, engine speed 501, J * dω/dt 503, the gearbox speed 504 and the corrected clutch ideal torque 505.

In phase 506 in which the ensuing engine torque 500 in constant a change of J * dω/dt 503 must be correlated with a change of the corrected clutch ideal torque when the corrected clutch torque 505 does not change. This condition is however fulfilled in most situations since the consumption as a rule does scarcely change short term. If these alterations are not correlated, ie a change of the corrected ideal clutch torque 505 causes no change of J*dω/dt (503), then the friction value must be corrected accordingly, if the change of 505 lies above that of 503 then the theoretical friction value must be lowered, because the actual friction value is less than the assumed value. If the reverse happens, it is necessary to proceed accordingly.

Through this method it is possible to calculate or determine the value of the friction value directly. It is

therefore possible at a time point when the engine speed gradient is zero, such as eg at position 507, to calculate the level of the value of the secondary consumer, since the engine torque is known. The following then applies:

$$M_{korr} = M_{an} - \frac{\mu_{ist}}{\mu_{theo}} * M_{ksollkorr}$$

Since the setting member lies between the calculated ideal torque $M_{Ksollkorr}$ 505 and the actual torque of the clutch 502, wherein the setting behaviour is generally not to be disregarded, it is possible to carry out a modelling of the setting member in order to increase further the quality of the adaptation in dynamic cases. With an electro-motor operated regulating unit of an electronic clutch management system, there is the possibility through the path measurement, for example in the transmitter cylinder, to calculate a theoretical actual torque 502 from the measured actual path and a characteristic line. This can be used in place of the ideal torque and is to be designated MK_{ist} 502. This thus includes the dynamic proportion which arises through the path regulation. The adaptation method is particularly advantageous in all driving conditions in which slip occurs. It is likewise advantageous that a division into a multiplicative and an additive proportion can be carried out.

A further possibility for adaptation offers the identification of the multiplicative proportion through the evaluation of starting off speeds. This simple possibility for identifying the additive and multiplicative proportion consists in the evaluation of a starting-off method. At the time point when the engine is idling at an idling speed, the driver has not provided any throttle, the moments applied by the engine are used for

the particular supply and compensation of the secondary assemblies. The value of the engine torque which is assumed in this situation can therefore be assumed as the stopping point for the value for the corrected torque. During the starting-off method when the driver provides throttle, the engine speed reached is evaluated at a certain time point. The engine speed is brought into connection with the adjoining clutch torque which is formed from the actual engine torque minus the engine torque shortly before providing throttle. From the table it is possible to compare whether the engine speed belonging to the adjoining engine torque coincides with the real actual engine speed. With larger deviations, a change exists in the friction value and the friction value existing in the control computer can then be corrected accordingly.

Figure 26 shows the adjoining engine torque 510 and the engine speed 511 as well as the gearbox input speed 512 as a function of the time. In front of a time point 517, the vehicle is in idling state wherein the secondary assemblies are evaluated with their power or torque take-up using the values in area 513. In the area after the time point 518 which is fixed after an acceleration phase, an ideal engine speed 514 can be determined from the value of the adjoining engine torque and this idling engine speed can be compared with the actual value 511 of the engine speed and thus an estimation can be carried out for the friction value. This method of procedure allows a division into a multiplicative and additive proportions wherein no effects appear in the case of a dynamic change in the setting member. The adaptation according to this method is particularly characterised in that this is only possible when starting off and a fault in the engine torque signal can influence the adaptation.

A further possibility of a method for adaptation can be designed in that the identification of the overall characteristic line is carried out using spot supporting points. This possibility, for systems with a detectable setting value, such as position of the disengagement system or disengagement path, can advantageously be carried out for the calculation if at the beginning of a dynamic adaptation the adaptive part, consumer moments and/or assembly losses are known approximately. A calculation of the offset signal in the case of unknown consumer moments and assembly losses could likewise be carried out wherein the determination can be undertaken through numerical processes.

In order to identify the characteristic line it would be necessary at certain path points or supporting points in the characteristic line to compare the corresponding calculated theoretical clutch torque 520 with that from the clutch characteristic line and the actual path 522. In the event of variation the supporting points would then be corrected incrementally, wherein the following applies:

$$M_{Kupplungtheo} = M_{An} - M_{korr} - J * d\omega/dt$$

Figure 27 shows from the actual value 522 in a time window 523 a change of the actual path of the setting member wherein the engine speed 524 and the gearbox 525 are detected. Using the supporting points 526 it is possible to determine from the actual path and the knowledge of the characteristic line of the torque transfer system the corresponding calculated clutch torque 520 which can be compared with the actual clutch torque. Figure 27 shows these values as a function of the time wherein the supporting points 526 can be defined using the position

details of the path of the setting member and the individual supporting points can be spread out according to the speed of the displacement of the setting member.

Figure 28 shows a clutch characteristic line 530 with supporting points 531 in which the clutch torque is determined and calculated. Furthermore the adaptation area 532 is illustrated which need not be fixed to the overall area of the clutch characteristic line wherein it can be advantageous if the torque area is adapted above a boundary value 533 and an adaptation below the boundary value 533 is carried out to the effect that a minimum value is set, as proposed for example in Figures 15a to 15e. Such an adaptation can be independent of the principle path recorded of the characteristic line wherein errors in the theoretical characteristic line are compensated.

The adaptation of the supporting points consequently also affects the operating areas which do not lie on the supporting points, but in these areas an extrapolation is necessary since the adapted operating points need not necessarily be used for starting off.

Figure 29a shows diagrammatically a drive train of a vehicle with a drive unit 600 and a torque transfer system 601 connected in the force flow on the output side of the drive unit. An automatic gearbox 610 is connected in on the output side of the torque transfer system, this automatic gearbox being illustrated diagrammatically as a cone pulley belt contact gearbox without restricting the generalities. The gearbox can also be an automatic infinitely adjustable gearbox, such as for example a friction wheel gearbox or a friction ring gearbox.

The cone pulley belt contact gearbox consists substantially of a variator which is comprised of two pairs of cone pulley sets 602a, 602b, 603a, 603b, and a contact means 604.

At least one fixed transmission step 605 is connected to the output side of the variator of the cone pulley belt contact gearbox and acts on a differential 606.

Figure 29b has the same structural arrangement except for the arrangement of the torque transfer system 611 which is connected in the force flow to the output side of the gearbox 610, such as a variator.

The contact pressure of the contact means is selected so that it does not lead to slipping of the contact means relative to the sets of cone pulleys. A control system controls the contact pressure of the contact means 604 between the pairs of cone pulleys in order to prevent slipping since slipping can lead locally to damage and even to destruction of the contact means.

With a change in the adjoining engine torque, an adaptive control can match or preset the transferable torque and a change in the operating point cannot result in slipping of the contact means, such as a chain.

The contact pressure of the contact means must be produced with an excess contact pressure in order in the event of for example torsion vibrations in the drive train to avoid slipping through a temporarily increased adjoining torque.

The control of the contact pressure with the lowest possible excess contact pressure is expedient since the excess contact pressure leads to friction losses and thus

to low performance and to increased fuel supply. A reduction in the excess contact pressure can lead to the danger of the contact means slipping through.

The fluctuations described above in the ensuing torque of the variator which is to be transferred can be calculated and taken into account by means of a control method since a dependence on the operating point can be adapted.

Furthermore, unforeseen torque shocks can occur on the output side, such as for example if the vehicle passes with turning tyres from a smooth road surface to a bumpy road surface. In this situation, a torque shock on the output side occurs which cannot be pre-calculated. Both the time path and the amplitude value cannot be calculated.

In order to protect the variator from such torque shocks according to Figures 29a, 29b a torque transfer system 601, 611 is mounted in the drive train and is controlled so that the torque transferable by the torque transfer system is always less than the torque transferable by variator.

The control of the transferable torque of the torque transfer system 601, 611 guarantees in each operating point that the transferable torque of the variator is greater than the transferable torque of the torque transfer system. The torque transfer system thus forms a moment-guided overload clutch which can be adaptively controlled at each operating point. Through the adaptive control of the torque transfer system it is possible to reduce the contact pressure of the contact means so that the safety reserves for protecting against slipping of the contact means are reduced. Thus the efficiency of the

gearbox can be increased without having to take safety risks regarding the variator.

The torque transfer system can be used as an actual safety clutch and/or as a turning set clutch and/or as a lock-up clutch of a torque converter or additionally as a clutch for adjusting the variator.

An arrangement of the torque transfer system on the output side is particularly advantageous since load shocks are recognised earlier from the output side than in the case of an arrangement on the drive side, since with the introduction of a torque the rotary masses of the variator still operate.

An arrangement on the output side furthermore has the advantage that when the vehicle is stationary and the engine is running the variator is in rotation and rapid adjustment and/or stationary adjustment can be rapidly carried out.

With an arrangement of the torque transfer system on the output side, it is necessary when determining and/or when calculating the ensuing engine torque to take into account the transmission of the variator and the losses.

The invention is not restricted to the embodiment illustrated and described but also includes variations which can be formed through a combination of features and elements described in conjunction with the present invention. Furthermore, individual features and methods of functioning described in connection with the drawings can be taken alone to represent an independent invention.

The applicant reserves the right to claim as being

essential to the invention further features disclosed up until now only in the description, more particularly in connection with the drawings. The patent claims filed with this application are thus only proposed wordings without prejudice for achieving further patent protection. This application is one of a series of applications, all based on application 9503372.6.

Application 9503372.6 describes and claims a method for controlling a torque transfer system with or without load distribution, more particularly for motor vehicles wherein the clutch torque which can be transferred from a drive side to an output side of the torque transfer system is used as the control value wherein this control value is calculated and/or determined in dependence on a drive torque.

Application 9803622.1 (Agents ref: P1507.P3C) describes and claims a torque transfer system for transferring torque from a drive side to an output side wherein on the drive side there is an internal combustion engine and on the output side there is a gearbox, and the torque transfer system has a clutch, a setting member and a control device.

Application 9803613.0 (Agents ref: P1507.P3D) describes and claims a monitoring method for a torque transfer system with a manually switchable gearbox wherein the shift lever positions and a drive torque of a drive unit on the drive side are detected by a sensor unit and at least one corresponding shift lever signal and at least one comparison signal are recorded and different possible characteristics of these signal paths are recognised and identified as a switching intention and then a switching intention signal is then supplied to a subordinate clutch

operating system.

Application 9803619.7 (Agents ref: P1507.P3E) describes and claims a method for controlling a torque transfer system with a device for controlling the torque transfer system, the torque transfer system is mounted secondary in the force flow of a drive unit and in front of and/or behind in the force flow of a translation-variable device, this translation-variable device is provided with a contact means which transfers a torque from a first means to a second means wherein the first means is in active connection with a gearbox input shaft and the second means is in active connection with a gearbox output shaft, the contact means is connected by contact pressure or tensioning in friction engagement with the first and the second means, and the contact pressure or tensioning of the contact means is controlled in dependence on the operating point, characterised in that the torque transfer system is controlled matching the torque, with a transferable torque which is dimensioned in each operating point so that the contact means of the translation-variable device does not start to slip.

CLAIMS

1. Control method for a torque transfer system with or without load distribution wherein the clutch torque which can be transferred from a drive side to an output side of the torque transfer system is used as the control value and this control value is controlled by means of a setting member which is provided with a setting value which is functionally dependent on the transferable clutch torque, so that the transferable clutch torque always lies within a pre-determinable tolerance band about the slip limit wherein this slip limit is then accurately reached when the action of a torque arising on the drive side exceeds the clutch torque transferable by the torque-transferring parts.

2. Control method as claimed in Claim 1, wherein a value is given to the setting member as the setting value which corresponds to the transferable clutch torque between the torque-transferring parts of the torque transfer system.

3. Control method as claimed in Claim 1 or Claim 2, wherein the setting value is determined in dependence on a transferable clutch torque and that in order to calculate this transferable clutch torque, a difference is formed from a drive torque value and a correction value wherein this correction value is increased or reduced in dependence on at least one condition value of the torque transfer system.

4. Control method as claimed in Claim 3, wherein the correction value is determined in dependence on a slip or different speed between a drive and an output speed, wherein the correction value is increased so long as the slip speed lies below a predetermined slip boundary value,

and the correction value is reduced so long as the slip speed lies above this value or another predetermined slip boundary value.

5. Control method as claimed in Claim 4, wherein the correction value is increased incrementally so long as the slip speed lies below one slip boundary value and the correction value is reduced step wise so long as the slip speed lies above the one or other slip boundary value wherein between the relevant stages there are stopping phases of adjustable connections within which the correction value is kept constant at the value set at the beginning of the relevant stopping phase.

6. Control method as claimed in any one of Claims 3 to 5, wherein the times in which the drive speed exceeds the output speed by a definite slip speed is recognised as the slip phase and that at the end of the relevant slip phase, the correction value is set to a definite value.

7. Control method as claimed in Claim 6, wherein the times in which the drive speed exceeds the output speed by a definite slip speed are recognised as the slip phases and that the correction value where the slip speed assumes its maximum value is stored in an intermediate memory and at the end of each relevant slip phase, the actual correction value is replaced by the stored correction value.

8. Control method as claimed in any one of Claims 3 to 7, wherein the correction value at the end of the relevant slip phase is kept constant at its relevant value for a fixable time length.

9. Control method as claimed in any one of Claims 3 to

8, wherein a starting value is given to the setting member in dependence on a characteristic field or a characteristic line which includes the area of all possible transferable clutch moments and has at least one partial area within which only one starting value for the setting member is allocated to each of the relevant transferable clutch moments.

10. Control method as claimed in any one of Claims 3 to 9, wherein to calculate the transferable clutch torque a difference is formed from a drive torque value and the correction value and that this difference is increased by a torque value dependent on slip.

11. Control method as claimed in any one of Claims 3 to 10, wherein the rise in the actual clutch torque is restricted in the form of a gradient restriction in that the actual value of the transferable clutch torque is compared with a comparison torque value which consists of a previously detected transferable clutch torque value and an additive fixable restriction value and that the relevant smaller torque value is given to the setting member as a new starting value in dependence on this comparison.

12. Control method as claimed in any preceding claim, wherein several condition values, such as eg the engine speed, throttle valve angle and/or the suction intake pressure, are detected from an internal combustion engine mounted on the drive side of the torque transfer system and that from these condition values, the drive torque of the combustion engine is detected by means of the stored characteristic line fields.

13. Control method as claimed in any one of Claims 3 to

7, wherein any possible load forks lying between the drive and the torque transfer system are monitored at least partially and/or at least at times and the measured values resulting therefrom are used to calculate the drive torque actually arising on the drive side of the torque transfer system.

14. Control method as claimed in any one of Claims 3 to 7, wherein a relevant part of the drive torque corresponding to a proportion factor is used to calculate the transferable clutch torque and that this proportion factor is determined each time by means of the stored characteristic line fields.

15. Control method as claimed in any preceding claim, wherein in the case of torque transfer systems without load distribution, such a load distribution is copied by a secondary control program.

16. Control method as claimed in any preceding claim, wherein measurable breakdown values, such as in particular temperatures and/or speeds are detected and are compensated at least partially through a parameter adaptation and/or through a system adaptation.

17. Control method as claimed in any preceding claim, wherein indirectly measurable breakdown values of the control method, such as more particularly ageing and mean variation of individual component parts of the torque transfer system are detected in that some condition values of the torque transfer system are monitored and the parameters actually disturbed are detected and corrected in dependence on this monitoring and/or virtual breakdown sources which can be switched on in the form of program modules are used in order to correct and/or compensate the

influence of the breakdown values.

18. Control method as claimed in any preceding claim, wherein a first engagement of the clutch is only possible after checking a user authorisation.

19. Control method as claimed in any preceding claim, wherein a user display is controlled in dependence on the status of the control method so that a switching command is given for the user.

20. Control method as claimed in any preceding claim, wherein stationary phases, more particularly of a vehicle, are detected by monitoring significant operating values, such as the accelerator pedal and/or gear stick position and/or tacho speed and when a definite time period is exceeded, the drive unit is stopped and is restarted when necessary.

21. Control method as claimed in any preceding claim, wherein operating phases of the torque transfer system with minimum or without load testing are recognised as freewheel phases and within these freewheel phases the clutch is opened and at the end of the freewheel phase the clutch is closed again.

22. Use of the control method as claimed in any preceding claim, to support an anti-locking system wherein when the anti-locking system responds, the clutch is completely disengaged.

23. Use of the control method as claimed in any preceding claim, to support an anti-slip control wherein the setting member is controlled in certain operating areas as claimed in the allowance of the anti-slip control.

24. Method as claimed in any preceding claim, wherein the clutch torque which can be transferred from a drive side to an output side of the torque transfer system is used as the control value wherein this control value is calculated and/or determined in dependence on a drive torque.

25. Method as claimed in any one of Claims 1 to 23, for controlling a torque transfer system with or without load distribution which controls the torque which can be transferred from a drive side to an output side of the torque transfer system, and comprises a sensor system for detecting the measured value and a central control or computer unit which can be connected therewith, wherein the torque which can be transferred by the torque transfer system is controlled so that the transferable torque is calculated, adapted and controlled as a function of a drive torque and deviations from the ideal state are compensated long term through corrections.

26. Method as claimed in any one of Claims 1 to 23, for controlling a torque transfer system, more particularly for motor vehicles which is connected in the force flow on the output side of a drive machine and is connected in the force flow in front of and/or after a device which can vary the transmission and which (process) controls the torque which can be transferred from a drive side to an output side of the torque transfer system, comprises a control or computer unit, which is in signal connection with sensors and/or other electronic units wherein the torque which can be transferred by the torque transfer system is controlled so that the torque which can be transferred is calculated and controlled adaptively as a function of a drive torque and deviations from the ideal state are compensated long term through corrections.

27. Method as claimed in any preceding claim, wherein the control value is controlled by means of a setting member which is provided at the input side with a setting value which is functionally dependent on the clutch torque which can be transferred, so that the clutch torque which can be transferred always lies within a predetermined tolerance band about a slip limit wherein this slip limit is then reached if the action of a torque arising on the drive side exceeds the clutch torque which can be transferred by the parts transferring the torque.

28. Method as claimed in any preceding claim, wherein the torque which can be transferred by a torque transfer system, such as a friction clutch and/or hydrodynamic torque converter with or without converter lock-up clutch and/or starting clutch for automatic gearboxes and/or turning set clutch and/or a torque transfer system connected at the input or output side of an infinite gearbox, such as an infinitely adjustable cone pulley belt contact gearbox is controlled as a function of a drive torque so that in the case of systems with load distribution, such as hydro dynamic torque converters with converter lock-up clutch, the torque which can be transferred by the clutch is detected as claimed in the torque equation

$$M_{KSoll} = K_{ME} * M_{AN}$$

$$\text{for } K_{ME} \leq 1$$

$$M_{Hydro} = (1 - K_{ME}) * M_{AN}$$

$$M_{KSoll} = K_{ME} * M_{AN}$$

$$\text{for } K_{ME} > 1$$

$$M_{Hydro} = 0$$

with

K_{ME} = Torque division factor
M_{KSoll} = Clutch Ideal Torque
M_{AN} = Adjoining torque
M_{Hydro} = Torque transferred by the hydro dynamic torque converter

and a torque difference between the torque M_{AN} adjoining the torque transfer system from the drive assembly and the torque M_{KSoll} which can be transferred by the clutch is transferred through the hydrodynamic torque converter wherein a minimum slip between the drive and output of the torque transfer system is independently set in dependence on the torque division factor K_{ME} and deviations from the ideal state are adaptively detected and compensated long term.

29. Method as claimed in any preceding claim, wherein the torque which can be transferred by the torque transfer system is controlled as a function of a drive torque so that in the case of systems without load distribution, such as friction clutch and/or starting clutch and/or turning set clutch and/or torque transfer system of an automatic gearbox or an infinitely adjustable cone pulley belt contact gearbox, the torque which can be transferred by the friction clutch or starting clutch is determined

$$M_{KSoll} = K_{ME} * M_{AN}$$

and a defined over pressing of the torque-transferring parts is carried out for $K_{ME} \geq 1$.

30. Method as claimed in any preceding claim, wherein the torque which can be transferred by the torque transfer system is controlled as a function of a drive torque so

that in the case of systems without load distribution, such as friction clutch and/or starting clutch and/or torque transfer system of an automatic gearbox and/or of an infinitely adjustable cone pulley belt contact gearbox, the torque which can be transferred by the torque transfer system is determined

$$M_{KSoll} = K_{ME} * M_{AN} + M_{Sicher}$$

and for $K_{ME} < 1$ a fictitious load distribution through a supporting control loop copies the behaviour of a hydrodynamic torque converter connected in parallel and a proportion of the transferable torque is controlled through the torque control and the remaining torque is controlled dependent on slip through a safety torque M_{Sicher}.

31. Method as claimed in Claim 30, wherein the safety torque M_{Sicher} is set in dependence on an operating point.

32. Method as claimed in Claim 30 or Claim 31, wherein a safety torque M_{Sicher} is detected and/or controlled in functional dependence on the slip Δn and/or the throttle valve position d as claimed in

$$M_{Sicher} = f(\Delta n,\ d).$$

33. Method as claimed in any one of Claims 30 to 32, wherein the safety torque M_{Sicher} is detected and/or controlled according to

$$M_{Sicher} = Const. * \Delta n.$$

34. Method as claimed in any preceding claim, wherein the torque division factor K_{ME} is constant over the entire

operating area of the drive train.

35. Method as claimed in any one of Claims 1 to 33, wherein the torque division factor K_{ME} assumes an individual value which can be detected from each operating point and/or assumes a constant value at least in a partial area of the operating area.

36. Method as claimed in any preceding claim, wherein the value of the torque division factor K_{ME} is in a functional connection dependent on the drive speed and/or the vehicle speed.

37. Method as claimed in any preceding claim, wherein the value of torque division factor K_{ME} depends only on the speed of the drive assembly.

38. Method as claimed in any preceding claim, wherein the value of the torque division factor K_{ME} is dependent at least in a partial area of the overall operating area both on the speed and on the torque of the drive assembly.

39. Method as claimed in any preceding claim, wherein the value of the torque division factor K_{ME} is dependent both on the output speed and on the torque of the drive assembly.

40. Method as claimed in any preceding claim, wherein at each time a specific ideal clutch torque is transferred by the torque transfer system.

41. Method as claimed in Claim 40, wherein the transferable clutch torque follows the ideal clutch torque.

42. Method as claimed in Claim 40, wherein the transferable clutch torque is matched taking into account a slight excess contact pressure ΔM lying in a scatter band in relation to the ideal clutch torque.

43. Method as claimed in Claim 42, wherein the excess contact pressure ΔM is dependent on the operating point.

44. Method as claimed in Claim 42, wherein the operating area is divided into partial areas and the contact pressure is fixed for each partial area.

45. Method as claimed in Claim 43, wherein the contact pressure and/or the transferable clutch torque is controlled variable in time.

46. Method as claimed in Claim 40, wherein the transferable clutch torque to be set does not understep a minimum value M_{Min}.

47. Method as claimed in Claim 46, wherein the minimum torque M_{Min} depends on the operating point and/or on the partial area of the operating area and/or on the time.

48. Method as claimed in any one of Claims 40 to 47, wherein the torque matching is carried out by means of a combination of a time-variable matching with a minimum value specific to the operating point.

49. Method as claimed in any preceding claim, wherein an operating point or an operating condition of a torque transfer system and/or of an internal combustion engine is determined from the condition values detected or calculated from the measured signals, such as in dependence on the engine speed and the throttle valve

angle, in dependence on the engine speed and the fuel through put, in dependence on the engine speed and the inlet manifold underpressure, in dependence on the engine speed and the injection time or in dependence on the temperature and/or the friction value and/or the slip and/or the load lever and/or the load lever gradient.

50. Method as claimed in any preceding claim, wherein in the case of a torque transfer system with internal combustion engine mounted on the drive side, the drive torque of the internal combustion engine is determined from at least one of the condition values of the operating point, such as engine speed, throttle valve angle, fuel throughput, inlet manifold underpressure, injection time or temperature.

51. Method as claimed in any preceding claim, wherein the torque $M_{AN} * K_{ME}$ adjoining the torque transfer system on the drive side is influenced and/or changed with a dependence which takes into account the dynamics of the system, wherein the dynamics of the system can be caused through the dynamic behaviour as a result of mass inertia moments and/or free angles and/or damping elements.

52. Method as claimed in any preceding claim, wherein means are provided which deliberately restrict and/or influence the dynamics of the system.

53. Method as claimed in Claim 51 or Claim 52, wherein the dynamics of the system are effected for influencing $M_{AN} * K_{ME}$ in a form of the gradient restriction.

54. Method as claimed in Claim 53, wherein the gradient restriction is effected as a limitation of a permissible increment.

55. Method as claimed in Claim 53, wherein the gradient restriction is produced in that the time change and/or the time-variable rise of a signal is compared with a maximum permitted slope or slope function and if the maximum permissible increment is exceeded, the signal is replaced by a replacement signal which is incremented with a previously defined slope.

56. Method as claimed in Claim 51, wherein the influence on the dynamics of the system is designed according to the principle of a time dynamic or variable filter wherein the characteristic time constants and/or amplifications are time variable and/or dependent on the operating point.

57. Method as claimed in Claim 54, wherein the dynamics of the system are considered and/or processed with a PT_1-filter.

58. Method as claimed in any one of Claims 51 to 57, wherein the dynamics of the system are marked by a maximum restriction.

59. Method as claimed in any one of Claims 51 to 58, wherein at least two means for influencing the dynamics of the system such as a gradient restriction and a filter stage are connected in series.

60. Method as claimed in any one of Claims 51 to 59, wherein at least two means for influencing the dynamics of the system such as a gradient restriction and a filter are connected in parallel.

61. Method as claimed in any preceding claim, wherein the dynamics of the internal combustion engine and the

dynamics of the secondary consumer are taken into consideration when determining the drive torque M_{AN}.

62. Method as claimed in Claim 61, wherein the mass inertia torque of the relevant flywheel masses and/or elements are used to take into account the dynamics of the internal combustion engine.

63. Method as claimed in Claim 61, wherein the injection behaviour of the internal combustion engine is used to take into account the dynamics of the internal combustion engine.

64. Method as claimed in any preceding claim, wherein deviations from the ideal state are compensated long-term by considering the secondary consumer and/or the correction and/or the compensation of faults and/or sources of breakdown.

65. Method as claimed in Claim 64, wherein the torque adjoining the torque transfer system on the input side is detected and/or calculated as a difference between the engine torque M_{Mot} and the sum of the torques of the secondary consumer taken up or branched off, wherein secondary consumer is to include at least substantially a climate control and/or a dynamo and/or a servo pump and/or a steering aid pump.

66. Method as claimed in Claim 65, wherein system condition values, such as the engine speed and the throttle valve angle, the engine speed and the fuel throughput, the engine speed and the inlet manifold underpressure, the engine speed and the injection time, the engine speed and the load lever, are used in order to determine the value of the engine torque M_{Mot}.

67. Method as claimed in Claim 66, wherein the engine torque M_{Mot} is detected from an engine characteristic field by means of the system condition values.

68. Method as claimed in Claim 67, wherein system condition values are used to determine the engine torque M_{Mot} and the engine torque is determined by solving at least one equation or an equation system.

69. Method as claimed in Claim 66, wherein the torque take-up of the secondary consumer is determined from the measured values, such as the voltage and/or current measured values of the dynamo and/or switching signals of the secondary consumer and/or other signals indicating the operating state of the secondary consumer.

70. Method as claimed in Claim 69, wherein the torque take up of the secondary consumer is determined by means of the measured values from the characteristic fields of the relevant secondary consumer.

71. Method as claimed in Claim 66, wherein the torque take-up of the secondary consumer is determined by solving at least one equation or an equation system.

72. Method as claimed in any preceding claim, wherein the corrected transferable clutch torque can be determined as claimed in the torque equation

$$M_{KSoll} = K_{ME} * (M_{AN} - M_{Korr}) + M_{Sicher}$$

and the correction torque $M_{(Korr)}$ is produced from a correction value which is dependent on the sum of the moments taken up by the secondary assemblies.

73. Method as claimed in any preceding claim, wherein a

correction is carried out of faults which have an effect on measurable system input values.

74. Method as claimed in any preceding claim, wherein measurable breakdown values are detected and/identified and are compensated and/or corrected at least partially through a parameter adaptation and/or through a system adaptation.

75. Method as claimed in any preceding claim, wherein measurable system input values are used to identify breakdown values and to correct and/or at least partially compensate same through parameter adaptation and/or system adaptation.

76. Method as claimed in any preceding claim, wherein system input values such as temperature, speeds, friction value and/or slip are used as values in order to identify a breakdown value and/or to correct and/or at least partially compensate same by means of a parameter adaptation and/or system adaptation.

77. Method as claimed in any one of Claims 73 to 76, wherein a compensation and/or correction of measurable breakdown values is carried out through an adaptation of the engine characteristic field.

78. Method as claimed in Claim 77, wherein a correction characteristic line field is produced from a comparison between the clutch ideal torque and actual torque and a correction value is determined for the relevant operating point which (correction value) is linked through addition with the value of the engine torque from the engine characteristic field.

79. Method as claimed in Claim 78, wherein analysis and/or measures are introduced in the light of a deviation detected in an operating point in order to calculate and/or fix the determinations and/or correction values in other operating points of the entire operating area.

80. Method as claimed in Claim 78, wherein analysis and/or measures are introduced in the light of a deviation detected in an operating point in order to calculate and/or fix deviations and/or correction values in other operating points of a restricted operating area.

81. Method as claimed in Claim 79, wherein the analysis and/or measures for determining and/or calculating the deviations and correction values in the further operating points take into account the entire or a restricted operating area.

82. Method as claimed in any one of Claims 79 to 81, wherein the analysis and/or measures for calculating the deviations and correction values in the further operating points only detect partial areas around the actual operating point.

83. Method as claimed in any one of Claims 79 to 82, wherein the analysis and/or measures for determining and/or calculating the deviations and/or correction values in the further operating points are carried out so that weighting factors evaluate or weigh up different areas of the entire operating area differently.

84. Method as claimed in Claim 83, wherein the weighting factors are selected and/or calculated as a function of the operating point.

85. Method as claimed in Claim 83 or Claim 84, wherein the weighting factors depend on the type of breakdown values and/or on the cause of the breakdown.

86. Method as claimed in any one of Claims 77 to 85, wherein after determining the correction value and/or after the weighting of the correction characteristic field, a time behaviour is imprinted on the correction value.

87. Method as claimed in Claim 86, wherein the time behaviour is determined through a beat frequency of a scanning of the correction values.

88. Method as claimed in Claim 86 or Claim 87, wherein the time behaviour is determined through at least a digital and/or analogue filter.

89. Method as claimed in any one of Claims 79 to 88, wherein the time behaviour is varied for different breakdown values and/or different sources of breakdown.

90. Method as claimed in any one of Claims 79 to 89, wherein the time behaviour is selected in dependence on the value of the corrections.

91. Method as claimed in any one of Claims 79 to 90, wherein the drive torque is adapted with an adaptation method with greater or smaller time constants than the time constants of the adaptation method of the clutch torque.

92. Method as claimed in Claim 87, wherein the time constants lie in an area from 1 second to 500 seconds, preferably however in an area from 10 sec to 60 sec and

more particularly preferably in an area from 20 sec to 40 sec.

93. Method as claimed in Claim 87, wherein the time constant is dependent on the operating point.

94. Method as claimed in Claim 87, wherein the time constant is selected or determined differently in various operating areas.

95. Method as claimed in any one of Claims 73 to 94, wherein a compensation and/or correction of measurable breakdown values is carried out through adaptation of the inverse transfer function of the transfer unit with setting member.

96. Method as claimed in any preceding claim, wherein indirectly measurable breakdown values, such as in particular the ageing and mean variation of individual component parts of the torque transfer system are detected in that some characteristic values of the torque transfer system are monitored and in dependence on this monitoring the parameters actually disturbed are detected and corrected and/or virtual breakdown sources which can be switched on in the form of program modules are used in order to correct and/or compensate the influence of the breakdown values.

97. Method as claimed in any preceding claim, wherein disturbances from the non-measurable influence values, such as the mean variation of individual component parts or ageing are detected through deviations from the condition levels of the system.

98. Method as claimed in any preceding claim, wherein

disturbances, such as mean variations or ageing or other non-measurable influence values are not detected from measurable input values and are only recognised by observing system reactions.

99. Method as claimed in any one of Claims 96 to 98, wherein the deviations from the condition values and/or observations of system reactions are measured directly and/or calculated in a method model from other measured values.

100. Method as claimed in Claim 99, wherein the detection of deviations from the calculated method models is carried out by means of reference characteristic fields and/or clear reference characteristic values of the system.

101. Method as claimed in any one of Claims 96 to 100, wherein in order to correct and/or compensate a detected disturbance from non-measurable input values a breakdown source is localised and/or a breakdown source is fixed and the deviations at these breakdown sources are corrected and/or compensated.

102. Method as claimed in any one of Claims 96 to 101, wherein in order to correct and/or compensate a recognised disturbance a fictitious breakdown source is fixed which need not be responsible for the disturbance at which the detected deviation is corrected.

103. Method as claimed in Claim 101 or Claim 102, wherein the fixed breakdown source is a function block which actually exists.

104. Method as claimed in Claim 101 or Claim 102, wherein the fixed breakdown source is a virtual breakdown model

whilst preserving the correcting action.

105. Method as claimed in any one of Claims 96 to 104, wherein the time path of the clutch actual torque is monitored and then analysed to see whether statements on the type of error and/or the detection of the breakdown source and/or the localisation of the breakdown source can be made.

106. Method as claimed in any preceding claim, wherein the adaptive correction of the breakdown values is carried out permanently.

107. Method as claimed in any preceding claim, wherein the adaptive correction of the breakdown values is only carried out in certain operating points and/or certain operating areas and/or time areas.

108. Method as claimed in any preceding claim, wherein the adaptation can also be active if the control is inactive.

109. Method as claimed in any preceding claim, wherein the adaptation is not carried out in special operating areas, such as in particular in the event of severe acceleration.

110. Method as claimed in Claim 109, wherein in the operating areas of the inactive adaptation the correction values of the breakdown values are used which were detected in operating areas of the active adaptation previously detected.

111. Method as claimed in Claim 109 or Claim 110, wherein in the operating areas of the inactive adaptation the correction values of the breakdown values are used which were extrapolated from correction values from previously

detected operating areas of active adaptation.

112. Method as claimed in any one of Claims 99 to 111, wherein virtual breakdown models and/or virtual breakdown sources are adapted for the area of the engine torque and/or for the area of the net engine torque after considering the secondary consumer and/or for the clutch ideal torque.

113. Method as claimed in any one of Claims 96 to 112, wherein the inverse transfer function of the transfer unit with setting member is used or applied as the virtual breakdown source.

114. Method as claimed in any preceding claim, wherein the engine characteristic field is used as the virtual breakdown source.

115. Method as claimed in any preceding claim, wherein virtual breakdown sources are used in order to define breakdown values whose original causes cannot be localised, such as eg mean variations in the area of the manufacturing tolerances of the individual component parts.

116. Torque transfer system for transferring torque from a drive side to an output side wherein on the drive side there is an internal combustion engine and on the output side there is a gearbox, and the torque transfer system has a clutch, a setting member and a control device and is operated in accordance with a method as claimed in any preceding claim.

117. Torque transfer system as claimed in Claim 116, for transferring torque from a drive side to an output side

wherein the torque transfer system is switched before or after on the output side in the force flow of a drive unit, such as internal combustion engine, and in the force flow of a translation-variable device, and the torque transfer system has a clutch and/or a torque converter with lock-up clutch and/or a starting clutch and/or a turning set clutch and/or a safety clutch restricting the transferable torque, a setting member and a control device.

118. Torque transfer system as claimed in Claim 116 or Claim 117, wherein the clutch is self-adjusting.

119. Torque transfer system as claimed in Claim 116 or Claim 117, wherein the clutch automatically adjusts to the wear on the friction linings.

120. Torque transfer system as claimed in any one of Claims 116 to 119, wherein for transferring the torque from a drive side to an output side the torque transfer system has a clutch, a setting member and a control device wherein the clutch is in active connection with the setting member through a hydraulic pipe which has a clutch receiver cylinder and the setting member is controlled by the control device.

121. Torque transfer system as claimed in any one of Claims 116 to 119, wherein to transfer torque from a drive side to an output side wherein a combustion engine is mounted on the drive side and a gearbox is mounted on the output side the torque transfer system has a clutch, a setting member and a control device wherein the clutch is in active connection with the setting member through a hydraulic pipe which has a clutch receiving cylinder and the setting member is controlled by the control device.

122. Torque transfer system as claimed in any one of Claims 116 to 121, wherein the setting member has an electric motor which acts through an eccentric on a hydraulic transmitter cylinder which is attached to the hydraulic pipe which is connected to the clutch, and that a clutch path sensor is mounted in the housing of the setting member.

123. Torque transfer system as claimed in Claim 122, wherein the electric motor, the eccentric, the transmitter cylinder, the clutch path sensor and the required control and power electronics are mounted inside a housing of the setting member.

124. Torque transfer system as claimed in Claim 123, wherein the axes of the electric motor and the transmitter cylinder run parallel to each other.

125. Torque transfer system as claimed in Claim 124, wherein the axes of the electric motor and the transmitter cylinder are set parallel to each other in two different planes and are in active connection through the eccentric.

126. Torque transfer system as claimed in Claim 124, wherein the axes of the electric motor runs parallel to a plane which is formed substantially by the plate of the control and power electronics.

127. Torque transfer system as claimed in any one of Claims 124 to 126, wherein a spring is mounted in the housing of the setting member concentric with the axis of the transmitter cylinder.

128. Torque transfer system as claimed in any one of

Claims 124 to 126, wherein a spring is mounted in the housing of the transmitter cylinder concentric with the axis of the transmitter cylinder.

129. Torque transfer system as claimed in Claim 127 or Claim 128, wherein a spring characteristic line of the spring is adapted so that the maximum force to be applied by the electric motor to disengage and engage the clutch is approximately the same size in the push and pull direction.

130. Torque transfer system as claimed in Claim 129, wherein the spring characteristic line of the spring is designed so that the resulting force path of the forces acting on the clutch is linear over the disengagement and engagement method of the clutch.

131. Torque transfer system as claimed in any one of Claims 116 to 130, wherein the electric motor acts with an engine output shaft through a worm on a segment wheel and a crank is attached to this segment wheel and is in active connection through a piston rod with the piston of the transmitter cylinder so that the push and pull forces can be transferred.

132. Torque transfer system as claimed in Claim 131, wherein the worm forms a self-locking gearbox with the segment wheel.

133. Method as claimed in any one of Claims 1 to 115, for a torque transfer system with a manually switchable gearbox wherein the shift lever positions and a drive torque of a drive unit on the drive side are detected by a sensor unit and at least one corresponding shift lever signal and at least one comparison signal are recorded and

different possible characteristics of these signal paths are recognised and identified as a switching intention and then a switching intention signal is then supplied to a subordinate clutch operating system.

134. Method as claimed in Claim 133, wherein at least one shift lever signal path is evaluated to detect the gear and this information is used to identify a switching intention.

135. Method as claimed in Claim 133 or Claim 134, wherein a shift lever signal and a comparison signal are evaluated so that intersecting points of these signal paths can be detected and then a switching intention signal is passed to a subordinate clutch operating system.

136. Method as claimed in any one of Claims 133 to 135, wherein with the manual gearbox a selection path is differentiated between the switching lanes and a switching path within the switching lanes wherein the switching path and/or the selection path is detected for determining the relevant gear lever position.

137. Method as claimed in any one of Claims 133 to 136, wherein the comparison signal is determined or formed from the gear lever signal wherein the gear lever signal is filtered, the filter signal thus produced is increased or reduced by an offset signal proportional with the relevant drive torque and the sum signal thus obtained is evaluated as the comparison signal.

138. Method as claimed in any one of Claims 133 to 137, wherein as soon as when evaluating the two signal paths of the gear lever signal and the comparison signal an intersection point is detected a switching intention

counter is set to a defined value and is counted up in dependence on a computer cycle and that a switching intention signal is sent to a subordinate clutch operating system when the switching intention counter has reached a definite counting value wherein the counting up of the switching intention counter can be stopped through a control signal.

139. Method as claimed in Claim 137 or Claim 138, wherein the gear lever signal can be filtered to form the filter signal with an adjustable delay time.

140. Method as claimed in Claim 137 or Claim 138, wherein the gear lever signal can be processed to form the filter signal with a filter with PT_1 - behaviour.

141. Method as claimed in any one of Claims 133 to 140, wherein the gear lever signal is monitored and a change in the switching path within a defined partial area of the gear lever path within a fixable measuring period is evaluated so that when a fixable switching path change threshold is understepped, a switching intention signal is sent to subordinate devices.

142. Method as claimed in Claim 141, wherein the measuring period is fixed so that it is always clearly greater than a half vibration period of the gear lever which is not operated during driving operation.

143. Method as claimed in Claim 141 or Claim 142, wherein the defined partial area of the gear lever path lies outside of the gear lever path areas within which the non-operated gear lever moves during driving operation.

144. Method as claimed in any one of Claims 141 to 143,

wherein the length of the measuring periods is fixed in dependence on the mean value formation of the gear lever vibration period.

145. Method as claimed in Claim 144, wherein it is determined whether the gear lever vibrates freely during the driving operation or more particularly by applying a hand has a vibration behaviour which is modified from this, and that the mean value formation for determining the length of the measuring period is carried out in dependence on the results of the monitoring.

146. Method as claimed in any one of Claims 141 to 145, wherein the direction of movement of the gear lever is detected and when this direction of movement is reversed, a control signal is sent to the switching intention counter and/or any switching intention signal which may be provided is rescinded.

147. Method as claimed in any one of Claims 133 to 146, wherein a constant value for forming the comparison signal is selected in dependence on the vibration amplitude which is typical of the operation of the non-operated gear lever of the torque transfer system.

148. Method as claimed in Claim 139, wherein the delay time with which the filter signal is formed is adapted to the vibration frequency of the gear lever which is not operated during driving operation.

149. Method as claimed in any one of Claims 133 to 140, wherein the drive load is monitored and on exceeding a fixable drive load, a control signal is passed further on to the switching intention counter.

150. Method as claimed in Claim 137, wherein the off-set signal is set in dependence on the relevant throttle valve angle of a combustion engine used as the drive unit.

151. Method as claimed in any one of Claims 133 to 150, wherein the switching and the selection paths of the gear lever are each determined by a potentiometer.

152. Method as claimed in any preceding claim, wherein the torque transfer system is mounted secondary in the force flow of a drive unit and in front of and/or behind in the force flow of a translation-variable device, this translation-variable device is provided with a contact means which transfers a torque from a first means to a second means wherein the first means is in active connection with a gearbox input shaft and the second means is in active connection with a gearbox output shaft, the contact means is connected by contact pressure or tensioning in friction engagement with the first and the second means, and the contact pressure or tensioning of the contact means is controlled in dependence on the operating point, wherein the torque transfer system is controlled matching the torque, with a transferable torque which is dimensioned in each operating point so that the contact means of the translation-variable device does not start to slip.

153. Method as claimed in Claim 152, wherein the contact pressure or tensioning of the contact means is determined and set in each operating point in dependence on the ensuing engine torque and/or the load distribution regarding the secondary consumer and an additional safety tolerance and the transferable torque of the torque transfer system is controlled in dependence on the operating point and the torque transferable by the torque

transfer system in the event of fluctuations in the torque leads to a slipping of the torque transfer system before the slip limit of the contact means is reached.

154. Method as claimed in Claim 152 or Claim 153, wherein the slip limit of the torque transfer system in each operating point is less or is controlled less than the slip limit of the contact means of the translation-variable device.

155. Method as claimed in any one of Claims 152 to 154, wherein the torque transfer system with its slip limit which is dependent on the operating point isolates and/or dampens torque fluctuations and torque impacts on the drive side and/or output side and protects the contact means against slipping through.

156. Method as claimed in any one of Claims 152 to 154, wherein the contact pressure or tensioning of the contact means is carried out in dependence on the operating point and a safety reserve is taken into consideration in addition to the ensuing torque and this safety reserve can be matched and/or adapted to the transferable torque as a result of the control of the transferable torque on the torque transfer system.

157. Method as claimed in Claim 156, wherein the safety reserve of the contact pressure or tensioning is kept as low as possible as a result of the slip protection of the torque transfer system.

158. Method as claimed in any one of Claims 152 to 157, wherein in the event of torque peaks the torque transfer system temporarily slips.

159. Device for carrying out the method as claimed in any one of Claims 152 to 158, wherein the translation-variable device is an infinitely adjustable gearbox.

160. Device as claimed in Claim 159, wherein the translation-variable device is an infinitely adjustable cone pulley belt contact gearbox.

161. Device as claimed in Claim 159 or Claim 160, wherein the torque transfer system is a friction clutch, a convertor lock-up clutch, a turning set clutch or a safety clutch.

162. Device as claimed in Claim 161, wherein the clutch is a dry or wet type of clutch.

163. Device as claimed in Claim 159, wherein a setting member which controls the transferable torque is provided which is controlled electrically and/or hydraulically and/or mechanically and/or pneumatically or the control of the setting member is carried out from a combination of these features.

164. Device as claimed in any one of Claims 159 to 163, wherein the device is equipped with at least one sensor for detecting a wheel speed and with means for detecting the engaged transmission of a gearbox wherein a central computer unit processes the sensor signals and calculates the gearbox input speed.

165. Device as claimed in Claim 164, wherein the detected wheel speeds are averaged and the gearbox input speed is determined or calculated from this mean signal by means of the translations in the drive train and by means of the gearbox transmission.

166. Device as claimed in Claim 164 or Claim 165, wherein one to four, preferably however 2 or 4 sensors are fitted for the detection of the wheel speeds.

167. Device as claimed in any one of Claims 164 to 166, wherein the sensors for detecting the wheel speeds are in signal connection with an anti-blocking system or are a component part of this anti-blocking system.

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Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.P): F2D (DCG), F2L (LT)

Int Cl (Ed.6): F16H 61/14; F16D 48/06

Other:

Documents considered to be relevant:

Category	Identity of document and relevant passage	Relevant to claims
A	US 4969545 (TOYOTA)	

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